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Industrial Electrical Engineering and Automation

Energy Study of Bucket Positioning Systems on Wheel Loaders

– Loader Linkages

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Abstract

As a stage in the development of the next generation of wheel loaders, all systems are evaluated in order to minimize the energy consumption. This master thesis aims to investigate the energy consumption for four linkages with different design. It will also investigate how the energy consumption changes when an electro-hydraulic compensation is added in order to keep the load parallel and what the total cost of ownership is. Work cycles for how the linkages should move during simulations were developed, both for the use of forks and for the use of a bucket. Models have been built using the software's *AMESim*, *MatLab* and *SimuLink*. The results from the simulations were analyzed and the energy consumption for the four different linkages was found. This results combined with manufacturing cost of the linkages resulted in a total cost of ownership (TCO) for the different linkages, based on the factors; work hours during the linkage´s lifetime and at what share it is driven with forks contra bucket. The result clearly showed that one of the linkages was favorable to use in a TCO perspective and that linkage was also recommended to use on future wheel loaders.

Som ett steg i utvecklingen av nästa generations hjullastare görs studier av olika delsystem för att se var energiförbrukningen ligger. Denna masteruppsats har som mål att ta reda på energiförbrukningen för fyra länkage med olika konstruktion. Den kommer även att undersöka hur energiförbrukningen förändras när en elektro-hydraulisk kompensering introduceras för att hålla lasten parallell samt vad totalkostnaden för de olika länkagen blir. Arbetscykler har tagits fram för hur länkagen ska röra sig under simuleringar, både för gaffelhantering och skophantering. Modeller har byggts med *AMESim*, *MatLab* och *SimuLink*. Resultaten från simuleringarna har analyserats och energiförbrukningen för de fyra olika länkagen fastställdes. Dessa resultat tillsammans med tillverkningskostnaden för länkagen ligger till grund för totalkostnadsanalysen som är en funktion av antal arbetstimmar och andel av tiden som länkaget används med gafflar kontra skopa. Resultatet visade tydligt att ett av länkagen hade en lägre totalkostnad. Rekommendation av vilket länkage som bör användas på framtida hjullastare baseras på totalkostnadsanalysen och därför rekommenderas det länkage som hade lägst totalkostnad.

Acknowledgements

I would like to thank Volvo CE for giving me the opportunity to conduct my thesis work at Volvo CE in Eskilstuna. I would like to thank the entire Hydraulic Department for answering my various questions. Special thanks are dedicated to my supervisors Robert Morelius and Magnus Bergström for helping me out, supporting me and giving me good advice all through my thesis. Also a special thanks to my supervisor at the division of Industrial Electrical Engineering and Automation at Lund University, Assoc. Prof. Ulf Jeppsson.

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Nomenclature

Notation	Unit	Description
$\boldsymbol{A}$	-	Matrix of force/torque components
$\boldsymbol{a_i}$	-	Acceleration vector of *i*
$\boldsymbol{B}$	-	Gravitational forces/torques
$\boldsymbol{C}$	-	Inertia forces/torques
$\boldsymbol{X}$	-	Vector of forces/torques
A_i	[m^2]	Area
C_{cyl}	[SEK]	Cost for the cylinders per kg
C_d	-	Orifice discharge coefficient
C_{Diesel}	[SEK]	Cost for one liter of diesel
$C_{h,f}$	[SEK]	Cost to use the linkage for one hour with a fork tool
$C_{h,b}$	[SEK]	Cost to use the linkage for one hour with a bucket
C_{Life}	[SEK]	Cost for a linkage during its lifetime
C_{manu}	[SEK]	Total cost for manufacturing a linkage
C_{mat}	[SEK]	Material cost for a linkage per kg
C_{proc}	[SEK]	Processing cost for a linkage per kg
d	[m]	Distance
D_p	[m^3]	Displacement of a pump
E	[J]	Energy
E_b	[J]	Energy needed for a bucket work cycle
E_f	[J]	Energy needed for a fork work cycle
F_c	[N]	Force of a cylinder
g	[m/s^2]	Gravitational acceleration
H_1	[N]	Force
H_2	[N]	Force
H_3	[N]	Force
H_4	[N]	Force
H_5	[N]	Force
H_6	[N]	Force
H_7	[N]	Force
M	[Nm]	Torque
m	[kg]	Mass
M_1	[Nm]	Torque
M_2	[Nm]	Torque
M_3	[Nm]	Torque
M_4	[Nm]	Torque
M_5	[Nm]	Torque
M_6	[Nm]	Torque
M_7	[Nm]	Torque
m_i	[kg]	Mass of link *i*
m_l	[kg]	Mass of a linkage
n	[RPS]	Number of rotations

$\lvert OA \rvert$	[m]	Distance between the points O and A
$\lvert OA_x \rvert$	[m]	The x-directed distance between the points O and A
P	[W]	Power
p	[Pa]	Pressure
q	[m^3/s]	Flow rate
R	[m]	Piston radius
r	[m]	Piston rod radius
R_c	-	Reynolds number
S_f	[%]	Proportion of the driving that is run with forks
t	[s]	Time
T_{Life}	[h]	Amount of hours that the linkage is used during a lifetime
v	[m/s]	Velocity
V_1	[N]	Force
V_2	[N]	Force
V_3	[N]	Force
V_4	[N]	Force
V_5	[N]	Force
V_6	[N]	Force
V_7	[N]	Force
Y_{Diesel}	[J/l]	Energy content in diesel
Δp	[Pa]	Change of pressure
ε	-	Percent of displacement $0 < \varepsilon \leq 1$
η_{Diesel}	-	Efficiency of a Diesel Engine
η_H	-	Mechanical hydraulic efficiency of a pump
η_V	-	Volumetric efficiency of a pump
μ	[Pa*s]	Dynamic viscosity
ρ	[kg/m^3]	Density
ω	[rad/s]	Rotational speed

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Chapter 1 – Introduction

Volvo Construction Equipment is a global company with products such as articulated haulers, excavators, wheel loaders, waste handlers and other equipment. Volvo CE is part of the Volvo Group "which is one of world´s leading manufacturers of heavy commercial vehicles and diesel engines" [1].

Volvo CE is about to develop a new wheel loader. With rising fuel costs and regulation of CO_2 emissions they want to know which of their loader linkages is most efficient. Loader linkage or linkage will in this report refer to the mechanism in front of the wheel loader on which it is possible to connect different tools, such as buckets, snow plows, timber grabbers and forks.

This thesis will investigate how linkages with different geometries will affect the energy consumption. It will also focus on the energy consumption and performance when an electro-hydraulic compensation is added for a better parallel movement when using forks for the different linkages.

There are several sizes of wheel loaders in production and this work will be conducted on one of the medium sized, L120 series. However the result can probably be applied to all sizes of machines, if desired.

1.1 Definitions

Here follows a few definitions needed to better understand the material.

VCE – Volvo Construction Equipment.

WL – Wheel Loader.

Hydraulic System – Fluid-based systems using liquids as transmission media are called hydraulic systems "from the Greek words hydra for water and aulos for a pipe, descriptions which imply fluids are water although oils are more commonly used" [2]. On a WL there are many things in the hydraulic system for instance; pumps, pipes, valves and cylinders.

Working Hydraulic System – Is the part of the hydraulic system that moves the linkage, including the linkage itself.

Tip Load – Is the amount of load that will make the wheel loader balance on the front wheel axle.

LS – Load Sensing.

Tilt angle – Same as Attachment Angle, the angle of the tool towards ground.

HPH – Hinge Point Height.

Load - Example of different kinds of loads are gravel, pallets, rocks and timber.

E-HCC – Electro-Hydraulic Compensated Case.

CCW – Counter Clock Wise.

TCO – Total Cost of Ownership.

ICE – Internal Combustion Engine.

1.1.1 Linkage description

The parts of a linkage are named as in Figure 1.1. All the bearings (or joints) on a linkage are numerated with a letter and can also be viewed in Figure 1.1. The most important ones are the O-bearing and the A-bearing, which will be used for defining the position of the tool. The O-bearing is the point on which the boom is attached to the front frame and the A-bearing is the point on which the bucket or other tool, is attached to the boom and will pivot around.

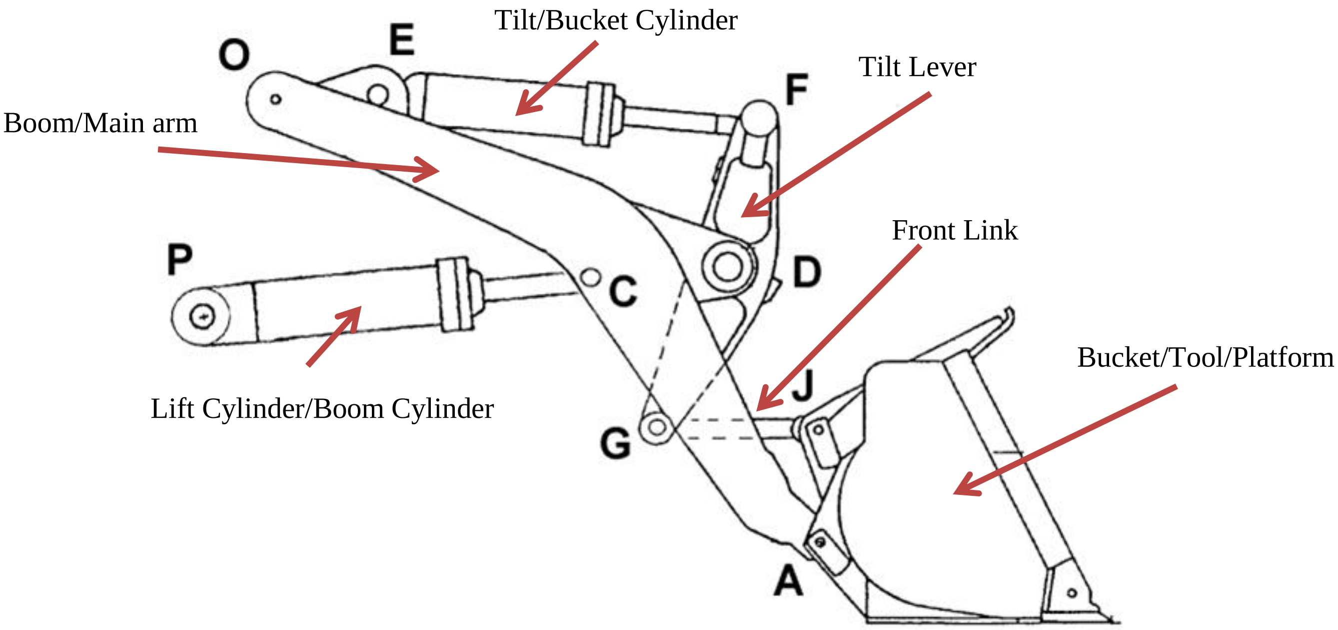

Figure 1.1 The nomenclature of a linkage and its bearings.

1.1.2 Linkage measurement points

Lift Height is the height of the A-bearing with origin in the O-bearing in y-direction as in Figure 1.2; it can be both positive and negative. The Tilt Angle is measured between ground and the bottom of the tool. Hinge Pin Height, HPH, is the distance between the ground and the A-bearing.

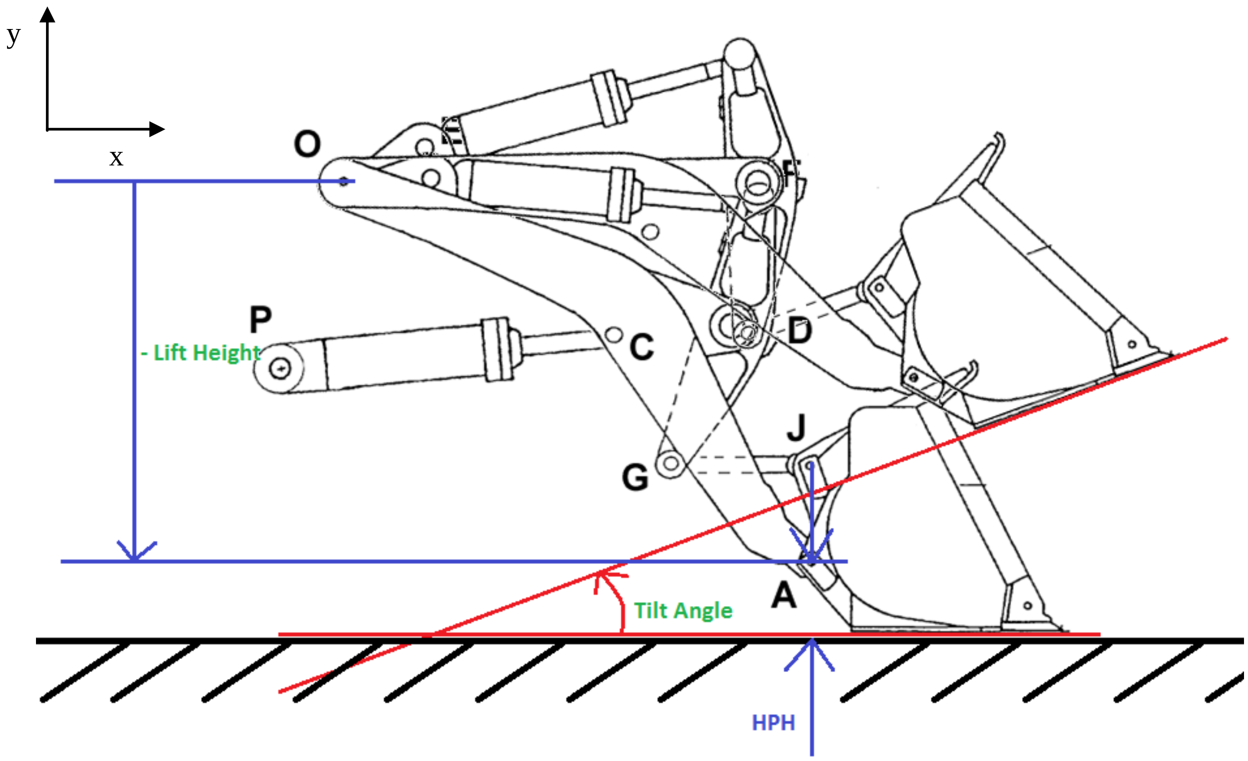

Figure 1.2 Linkage measurement points.

1.1.3 Parallel movement

The simplest linkages consist of only a boom and a lift cylinder as in Figure 1.4. It is clear that such a linkage is not of much use for anyone since the load can only be lifted up and put down. It also limits what kind of load that is possible to lift, without some of it falling off. The reason for that is that the tilt angle is coupled with the lift height and will therefore change with it. Therefore all linkages used on WL are equipped with a tilt cylinder enabling the driver to control the tilt angle.

The linkage can be either stiff or self-adjusting. Stiff means that the tilt angle will change the same as the boom´s rotation around the O-bearing, when only the lift cylinder is used. A self- adjusting linkage will have a geometry that changes the tilt angle less (or more) than the rotation of the boom around the O-bearing as shown in Figure 1.3. It is most common to have a self-adjusting linkage.

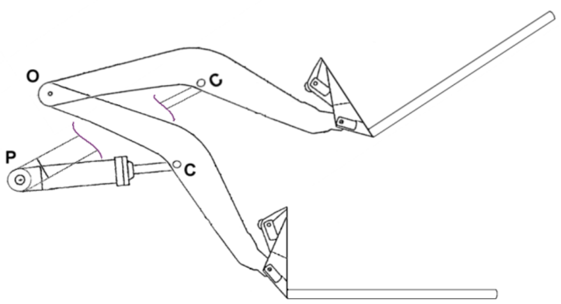

Figure 1.4 An example of a stiff linkage without tilt cylinder is shown.

Figure 1.3 An example of a self-adjusting linkage [7].

When designing the geometry of a linkage it is important to take into account how the desired self-adjustment should behave. The purpose of having a self-adjusting linkage is that the driver has one thing less to worry about when working. It is desired that the load stays parallel when lifting and lowering the load. However, parallel movement can be measured in different ways.

When using a bucket, parallel movement is considered when the bucket has tilted back, positive tilt angle, and the driver is about to lift the linkage as the bucket in Figure 1.5.

When using forks to lift pallets another angle is desired so the forks are horizontal all the time, placing the tilt cylinder and tilt lever in another position compared to the case with the bucket, see Figure 1.6. It is more important that the forks stays parallel at all time than the bucket. The bucket can vary a few degrees without the problem of spilling the load out.

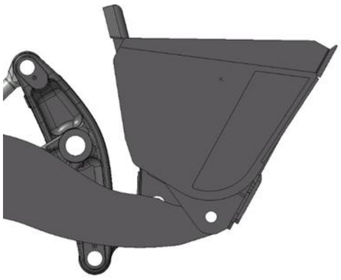

Figure 1.5 Example of when the tilt cylinder and tilt lever has tilted back [9].

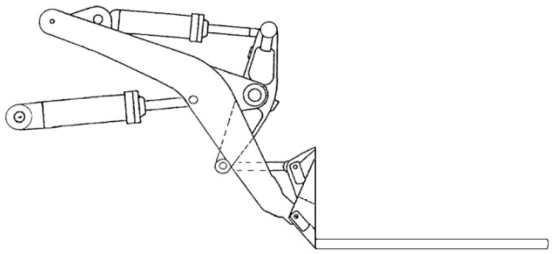

Figure 1.6 Example of when the tilt cylinder keeps the tool in a plane position.

The design of a linkage will have a big effect on how the parallel movement will be, an illustration of how differently two linkages behave is shown in Figure 1.7 for three cases; tilted in, tilted out, and plane bucket. Plane bucket is the position when the bucket or other tool has a tilt angle that is zero and when HPH is at its lowest position, as in Figure 1.6.

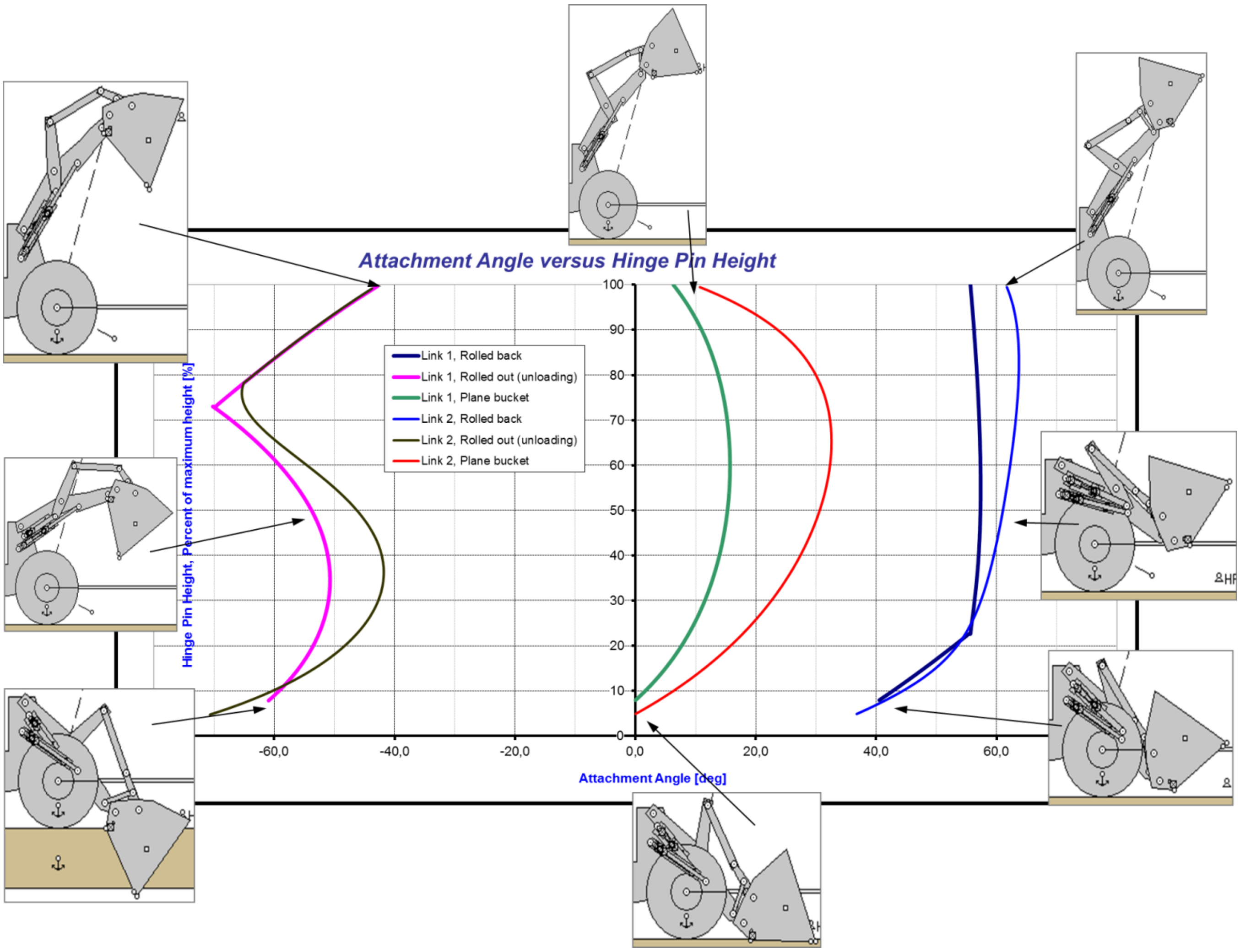

Figure 1.7 Parallel movement of two different linkages, AA versus HPH.

The self-adjustment is not perfect for either of the two linkages shown but is clearly optimized for a certain angle of the tool.

1.2 Problem statement

This report will investigate the following tasks for four different linkages:

1. Compare different mechanisms when it comes to parallel movement, operability.
2. Compare the weight of the constructions.
3. Compare energy consumption of the different mechanisms. Use of a “standard” lift-cycle from ground to a certain height.
4. Compare the energy consumption when an electro-hydraulic compensation is used (on the tilt cylinder) to keep the load parallel.
5. Cost calculation for the different linkages (for production and) for a TCO as cost per hour.
6. Suggest other ideas of linkages, if found.

The third and fourth tasks has no research done before and will be the main target of this report. The goal is to recommend a mechanism for a wheel loader, based on the parameters; energy consumption, cost, operability, performance, and weight.

1.3 Approach and outline

The approach will be to study the hydraulic system and linkages of WL L120. Build models of them using the software *AMESim*, *MatLab*, and *SimuLink*. Decide what constraints that are relevant for the thesis and specify the motion that the linkage should follow during the simulations. Run the simulations and analyze the results.

Some data in the tables, diagrams and in the results will be normed based on the wish of Volvo CE.

1.4 Related work

There has been some investigations of the usefulness and correctness of modeling hydraulics and mechanics in a case of a prosthetic arm. The conclusion where that a model built with commercial software can approximate quite well how the real prosthesis behave, but not exact [3]. It is to assume that different software on the market will behave similar to each other when it comes to building models of hydraulic systems.

Research about optimizing an existing linkage position of joints using kinematic and dynamic analysis has been conducted. The analysis are based on a work cycle that specifies the Horizontal and Vertical digging force needed versus time [4]. However it does not investigate what type of linkage would be best to start off with nor if the linkage behaves as it is supposed to.

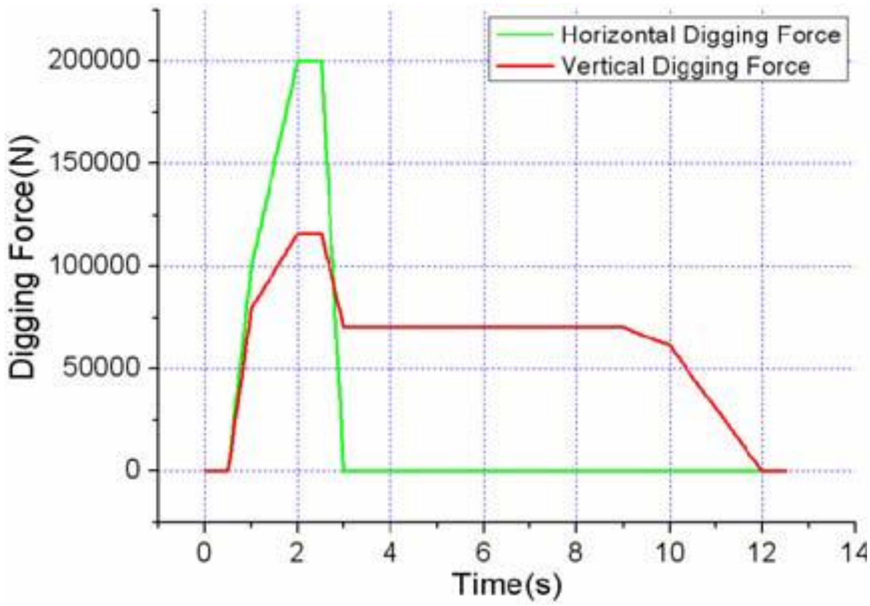

Figure 1.8 Force reference used in [4].

Research has been done about the most efficient way of lifting and emptying a bucket [5] and basically it is better not to lift the center of mass higher than necessary, making the bucket "roll" when it is emptying the bucket. The work focuses on how to use an existing linkage optimally, how to move the bucket instead of deciding what linkage type is best to use.

I have not found any other research about which design of a linkage that gives the lowest energy consumption.

Chapter 2 – Basic Hydraulics

On a WL there are many functions that are driven by the hydraulic system, brakes, steering, cooling system and loader linkage. Some hydraulic parts and concepts are briefly introduced here.

2.1 Cylinders

A cylinder is a linear actuator that will transform fluid flow and pressure into a linear movement and a force. A cylinder has a plus side and a minus side as shown in Figure 2.1. There are cylinders that only can be controlled in one direction, single-acting, often used when gravity or a spring pushes it back, and there are cylinders that can be controlled in both directions, double-acting. On a WL linkage the cylinders are double-acting since there is a need of an acting force in both directions.

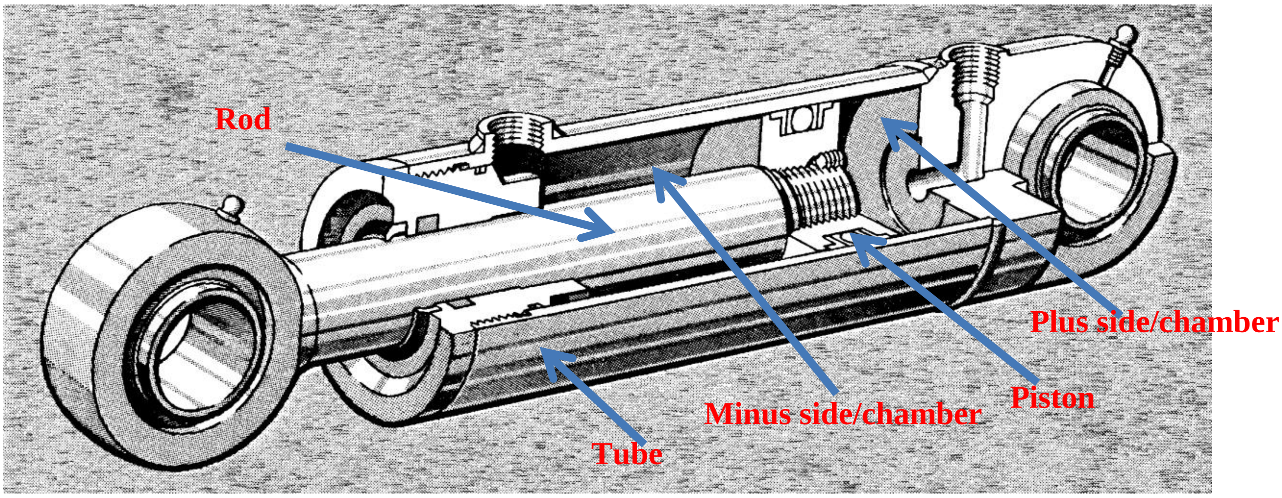

Figure 2.1 Cross section of a cylinder with names of the different parts [13].

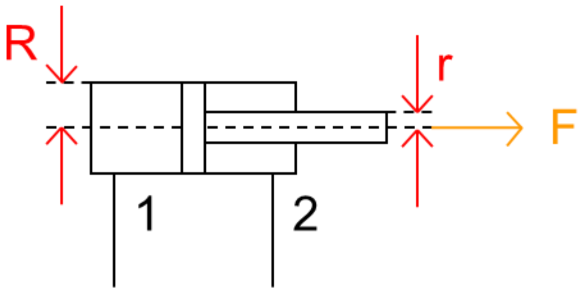

Figure 2.2 Notations of a cylinder.

The force F in Figure 2.2 that will be exerted by a cylinder can be calculated through the pressure in the system and the area of the cylinder. The piston rod will use some of the area that the fluid can push on and therefore different pressures on the two sides will be needed in order to keep the rod still. When fluid is pushed into the cylinder through port 1 the force becomes [2]:

$$F_c = p \cdot A_1 \tag{2.1}$$

where p is the pressure and A is the area:

$$A_1 = \pi \cdot R^2 \tag{2.2}$$

When retracting the cylinder the force will be:

$$F_c = -p \cdot A_2 \tag{2.3}$$

with a smaller area:

$$A_2 = \pi(R - r)^2 \tag{2.4}$$

The fact that a lower pressure is needed to extend than to retract the cylinder with the same force, comes into great use when designing how the cylinder should be placed in a linkage. It is preferable that the most demanding movement should use the force from extending the cylinder since it is greater for the same pressure in the system. The most demanding movement is when the bucket is being filled in the gravel pile and tilts in (increasing the tilt angle). If a linkage is configured so that the tilt cylinder uses the retract motion to tilt in, the cylinder is said to be mounted overlying.

The velocity v of a cylinder does not depend on the pressure in a system but rather on the fluid flow and the cylinder area. The speed can be expressed as distance, d, divided by time, t.

$$v = \frac{d}{t} \tag{2.5}$$

The flow rate can be expressed as [2]:

$$q = A \cdot v \tag{2.6}$$

The velocity of a cylinder can be calculated by knowing its area and what flow the cylinder is provided with.

2.2 Valves

A valve is a way to control the flow through a pipe. It can be opened fully, partially or closed. There are many types and variants of valves [2] that can open and close more than one pipe flow at a time. As seen in Figure 2.3 the valve has three positions and it controls the flow for two pipes. The flow through a valve can be looked upon as through an orifice and can be calculated by Equation 2.7 [6].

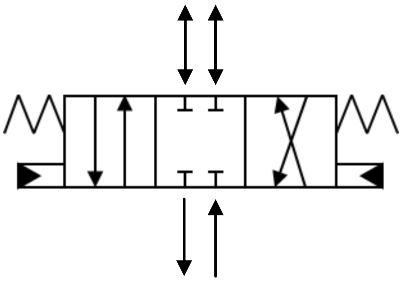

Figure 2.3 A 3-Position-2-Port valve.

$$q = C_d \cdot A \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}} \qquad 2.7$$

A check valve is shown in Figure 2.4. A check valve only lets the flow pass from side 1 to side 2 and only if the pressure at side 1 is higher than the pressure at side 2. More valves and other hydraulic components can be found in Appendix A.

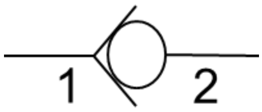

Figure 2.4 A check valve.

2.3 Pumps

The pump is the part of the hydraulic system that supplies the system with fluid flow and a pressure. It takes fluid from a tank and pushes it into the pipe system. When there is a resistance in the system the flow will build up a pressure, which could for instance be used to create a force on a cylinder.

There are many different types of pumps available, for a thorough brief of pumps A. Parr [2] is recommended. A pump can either have a fixed volume that it pushes for every rotation or it can be variable. Variable pumps are used today for many mobile hydraulic applications because it can lower the fuel consumption when the maximum flow is not required. It is however a more expensive solution than having fixed volume pumps.

For an ideal pump the displacement for the pump can be calculated as:

$$D_p = A_k \cdot R \cdot 2\pi \tag{2.8}$$

where A_k is the marked area in Figure 2.5, R is the radius and 2π corresponds to a whole revolution.

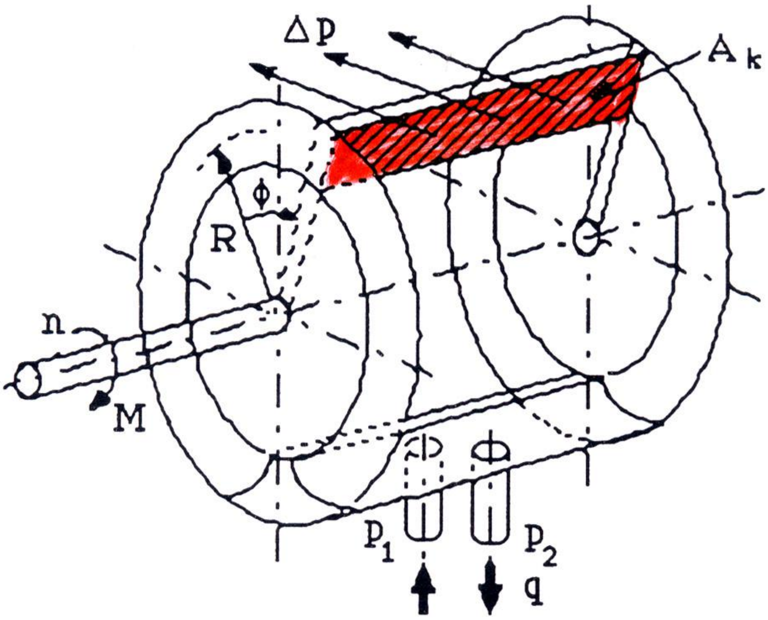

Figure 2.5 The area used in pump calculations is marked with A_k [13].

If the pump has a variable displacement then the instant displacement can be described as:

$$\varepsilon \cdot D_p \tag{2.9}$$

with ε as $0 < \varepsilon \leq 1$, for a pump that only can revolve in one direction. The flow can be described as the displacement multiplied by the number of revolutions per second.

$$q = \varepsilon \cdot D_p \cdot n \tag{2.10}$$

The torque needed to be applied is stated by Equation 2.11.

$$M = A_k \cdot R \cdot \Delta p = \frac{\varepsilon \cdot D_p}{2\pi} \cdot \Delta p \tag{2.11}$$

A real pump has losses from leakage and friction. The friction affects the torque and the leakage affects the flow. By adding the volumetric efficiency and the hydro-mechanical efficiency to Equations 2.10 and 2.11 the flow and torque become:

$$q = \varepsilon \cdot D_p \cdot n \cdot \eta_V \tag{2.12}$$

$$M = \frac{\varepsilon \cdot D_p}{2\pi} \cdot \Delta p \cdot \frac{1}{\eta_H} \tag{2.13}$$

where η_V is the volumetric efficiency and η_H is the hydro-mechanical efficiency. The power needed for the pump can be calculated by [2] Equation 2.14.

$$P = M \cdot \omega = p \cdot q \tag{2.14}$$

Typical efficiency maps of a pump can be found in Figure 2.6 and Figure 2.7 as a function of pressure and rotational speed. As can be viewed the mechanical efficiency will have a much bigger impact on the energy consumption than the volumetric efficiency. The mechanical efficiency varies a lot, between 20-80%. The volumetric efficiency is almost constant and lies around 97 %.

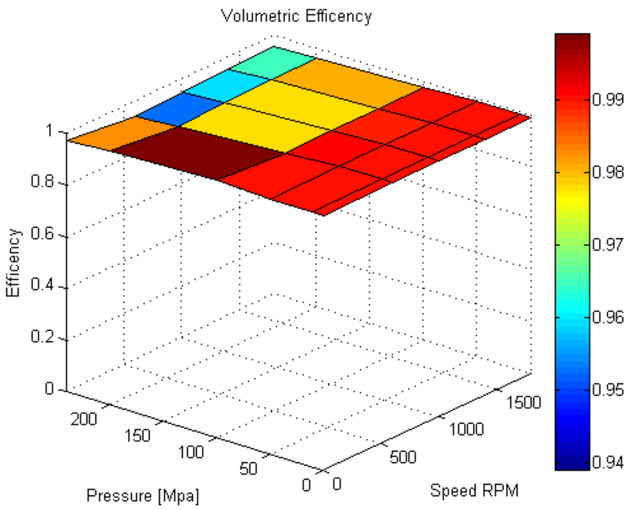

Figure 2.7 A typical volumetric efficiency map of a fixed pump.

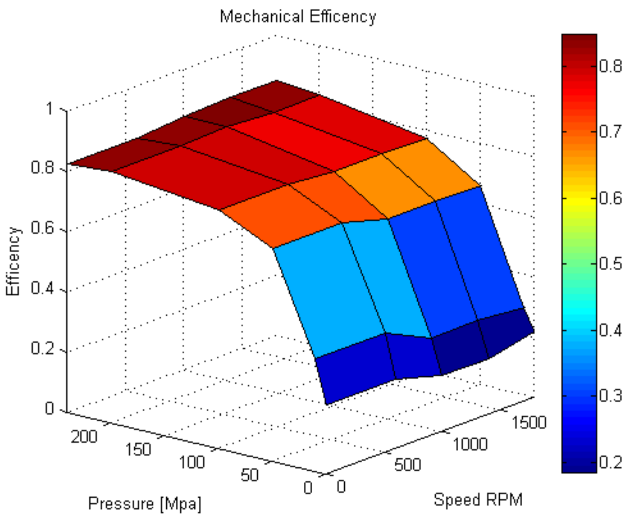

Figure 2.6 A typical mechanical efficiency map of a fixed pump.

2.4 Fluid flow

It is the fluid flow from the pump that supplies the hydraulic system. Fluid flow can be measured in different ways, *volumetric fluid flow*, *mass flow* and *velocity of flow* [2]. The fluid flow can be laminar or turbulent depending on what speed and fluid it is. A way of finding out if the flow is laminar or turbulent is to calculate the Reynolds number, R_c:

$$R_c = \frac{v \cdot d \cdot \rho}{\mu} \tag{2.15}$$

where v is the flow velocity, d is the diameter of the pipe, ρ is the fluid density and μ is the viscosity. A good rule of thumb is that the flow is laminar if $R_c < 2000$ and turbulent if $R_c > 10000$ [2]. In between both laminar and turbulent flows can occur.

2.4.1 Cavitation

Cavitation is a phenomenon where vapor bubbles are being created and popped. It often happens when a fast change of pressure occurs. It often leads to wear of pipes and valves in a hydraulic system and needs to be avoided [2].

2.5 Load sensing

A load sensing (LS) system is used to control the pump flow in order to minimize the energy consumption, no fluid will be pumped unless it is needed. An LS-system usually contains a variable displacement pump and uses the fact that a pressure drop, Δp, across a valve changes when the load changes. The pump will try to maintain a constant $\Delta p = C$.

An example of how a load sensing system is built up can be viewed in Figure 2.8. The pump is supplied with a pressure signal, the red dotted line, containing the highest pressure in the system. The highest load in the system gives rise to the highest pressure. When one or both the main valves is activated the pressure across that valve or valves will be used, otherwise the load sensing pressure (LSP) will be zero. Imagine that both cylinders in Figure 2.8 will be exposed to external forces with $F_2 > F_1$ and that the valves will be activated as the blue arrows show. Then the pressure on the plus side of *cylinder 1* will be higher than on the minus side. *Check valve 1* will let the highest pressure pass by i.e. from the plus side. The same thing will happened for *cylinder 2* and *check valve 2*. *Check valve 3* will let through the highest pressure, which in this case will be from *check valve 2,* since $F_2 > F_1$. The LSP is then led to the pump. The pump will try to maintain the pressure drop Δp as $PP - LSP = C$. PP is the pump pressure. When $\Delta p > C$ the pump will decrease the displacement i.e. reduce the flow and pressure. When $\Delta p < C$ the pump will increase the displacement of the pump.

If F_2suddenly would become zero then *check valve 3* would let through the pressure from check *valve 1*. This would be a smaller pressure than before and $\Delta p > C$ resulting in that the pump would decrease the displacement.

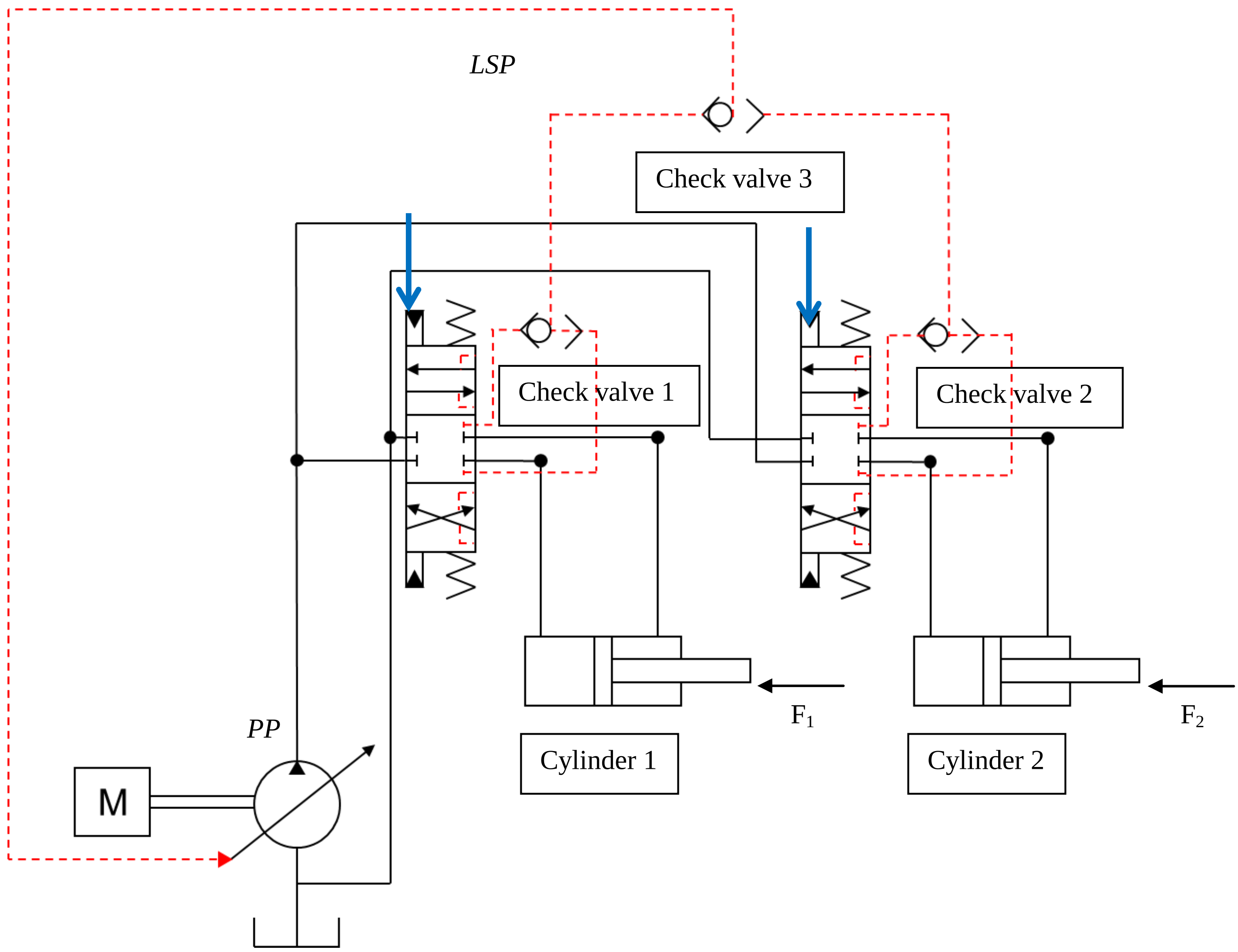

Figure 2.8 A load sensing system.

Chapter 3 – Wheel Loader, L120 Series

Figure 3.1 A Volvo Wheel Loader of the L120 series [7].

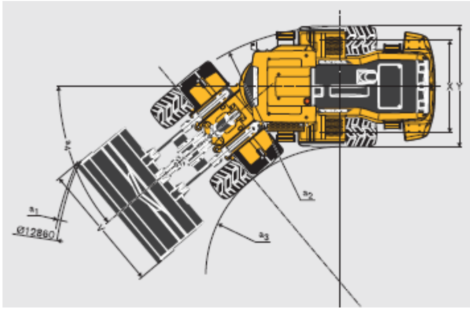

Figure 3.2 The maximum turn position of a WL [7].

The L120 series of WL, Figure 3.1, is mostly used as a multipurpose vehicle since it is big enough to be used to fill trucks with gravel but small enough to be legal to drive on roads. It can also be used for timber loading, snow plowing and pallet loading. Since it is used in many sensitive situations, narrow areas and pallet loading, it needs to be equipped with a good parallel movement. Today the TP-linkage is standard equipment on this model.

Compared to a regular car, on which the gas pedal controls the torque, on a wheel loader the gas pedal controls the engine speed (RPM). It is said that the vehicle is engine speed controlled not momentum controlled. This difference has a big impact on how it feels to drive a WL [5].

3.1 Dimensions and specifications

In order to get some perspective of the size of the machine some data is presented in Figure 3.3.

		Standard boom
B	mm (in)	6580 (259)
C	mm (in)	3200 (126)
D	mm (in)	440 (17.3)
F	mm (in)	3380 (133)
G	mm (in)	2132 (83.9)
J	mm (in)	3770 (149.2)
K	mm (in)	4100 (161.4)
O	°	54
P_{max}	°	51
R	°	42
R_1*	°	47
S	°	67
T	mm (in)	104 (40.1)
U	mm (in)	510 (20.1)
X	mm (in)	2070 (81.5)
Y	mm (in)	2670 (105.1)
Z	mm (in)	3340 (131.5)
a_2	mm (in)	5730 (225.6)
a_3	mm (in)	3060 (120.5)
a_4	±°	40

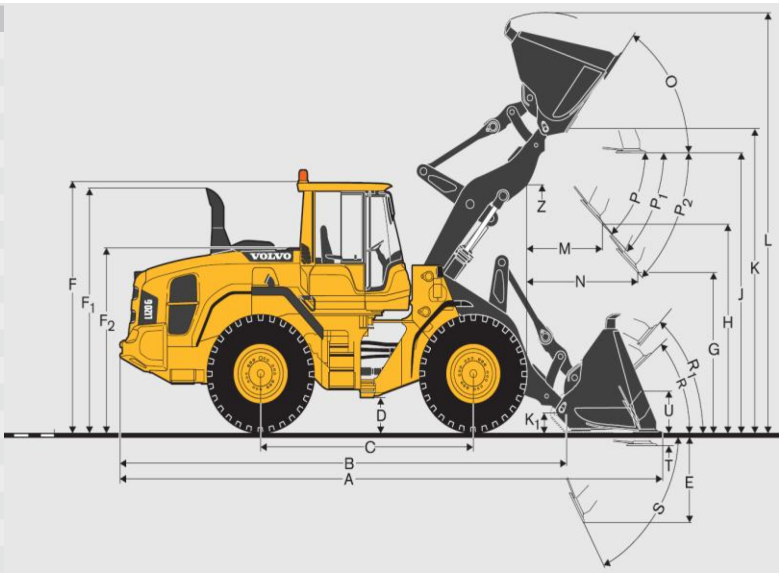

Figure 3.3 Dimensions of a Volvo L120 Wheel Loader [7].

The mass of the vehicle varies depending on what tools and wheels that are mounted but is usually just below 20,000 kg. It is capable of lifting about 14,000 kg when standing straight and 12,000 kg when turned maximum as in Figure 3.2. That is what the name L120 stands for, the maximum tipload when turned. The usual work load is about half the tipload i.e. 6,000 kg.

3.2 Linkages

When designing a linkage there are several factors that need to be taken into account. Except the self-adjustment also the breakout torque (at all positions), the tip load and other parameters have to be accounted for. Breakout torque is the torque that can be provided on the bucket (or other tool) around the A-bearing, as shown in Figure 3.4. It is especially important when filling a bucket at ground and emptying it at top position, since that requires a high torque. The breakout torque varies a lot for a linkage depending on what position it currently is at. An example of how the breakout torque changes with the attachment angle (AA) and hinge point height (HPH) for two linkages with different geometries is shown in Figure 3.5. As can be viewed the breakout torque can vary a lot.

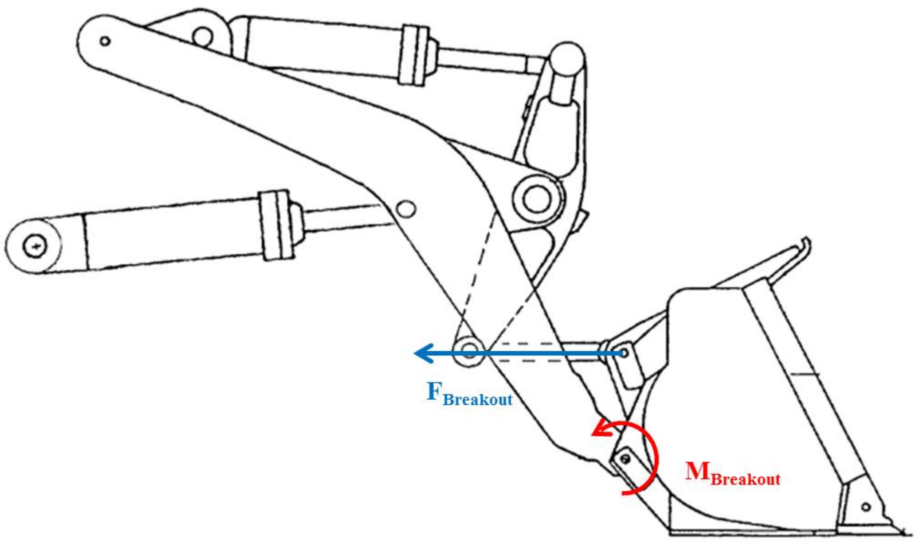

Figure 3.4 Breakout torque and force displayed.

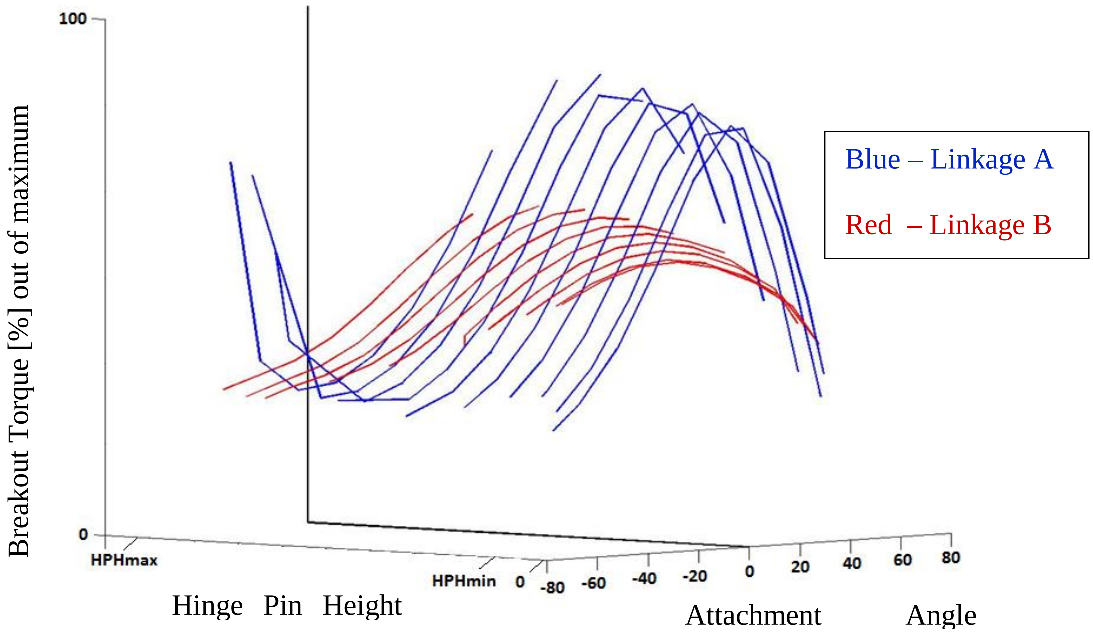

Figure 3.5 Breakout torque versus HPH versus AA.

As mentioned before there is a preferable way of positioning the tilt cylinder, however it is often hard to fit the tilt cylinder where it would be best to have it, the boom is usually in the way as shown in Figure 3.6 and Figure 3.7. This often results in a wide linkage that partially will block the view of the driver when raised. All linkages need to compromise between these and many more factors.

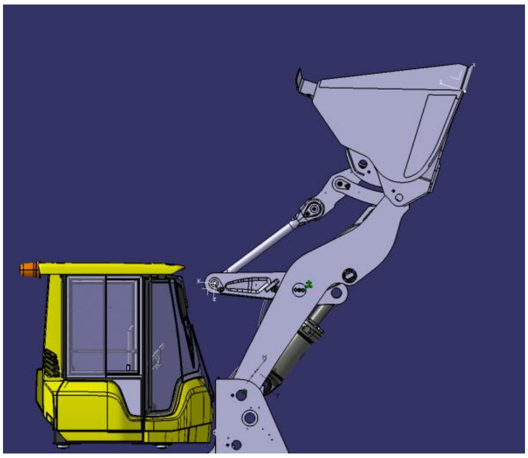

Figure 3.6 A Linkage in an upper position, blocking the view for the driver. Shown from the side.

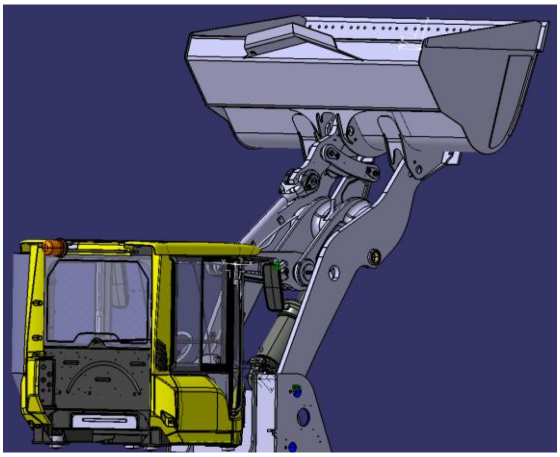

Figure 3.7 A Linkage in an upper position, blocking the view for the driver. Shown in 3-D from behind

3.2.1 TP-linkage

The TP-linkage, Figure 3.8, is an old Volvo invention [7] and is self-adjusting with a good parallel motion. TP stands for *Torque Parallel*. A TP-linkage has a breakout torque that does not vary much versus HPH or AA. This can be viewed in Figure 3.5 as linkage B.

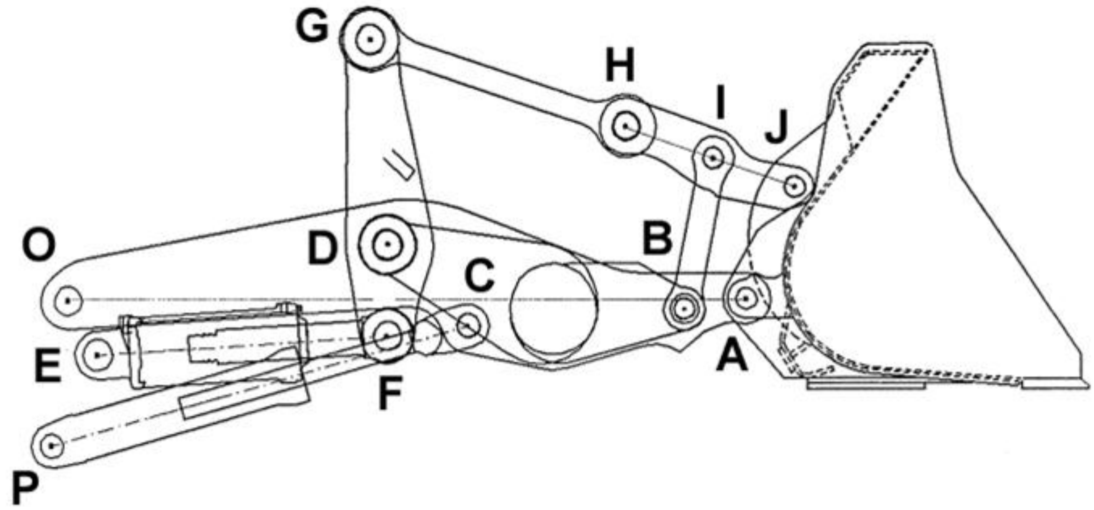

Figure 3.8 Example of a TP-linkage.

3.2.2 Z-linkage

The Z-linkage is the most used linkage today among all manufactures. Its geometry is shown in Figure 3.9. It is called Z since it has the shape of a z. It is self-adjusting, but not as good as the TP linkage. It is better at handling heavy tip loads because the mass center of the load is closer to the front axle than that of the TP. The breakout torque for a Z-linkage is high at bottom position and low at top position for a plane bucket, see linkage A in Figure 3.5.

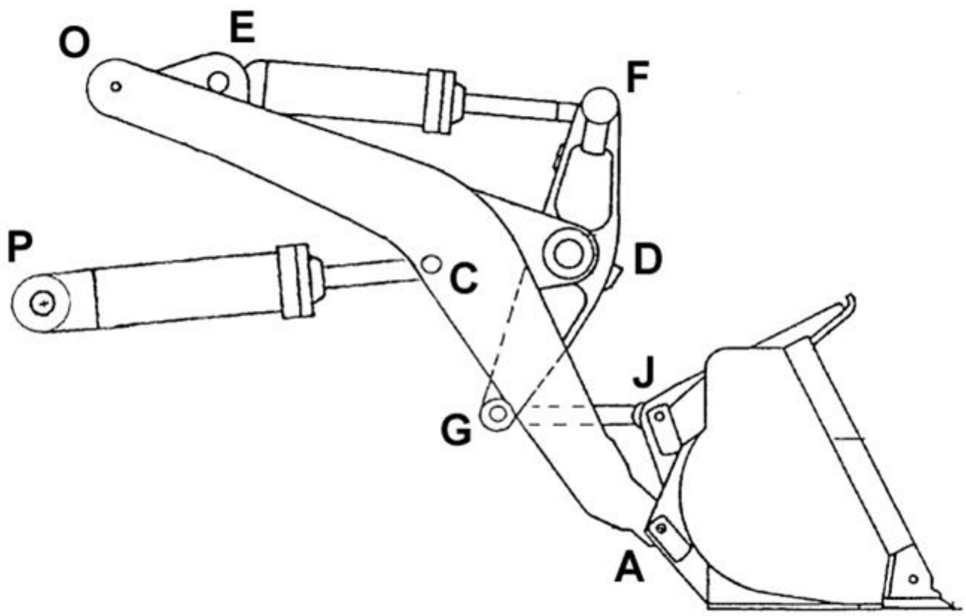

Figure 3.9 Example of a Z-Linkage.

3.2.3 TBM-linkage

The TBM-linkage, Figure 3.10, is potentially good at parallel adjustment but has an overlying tilt cylinder, reducing its break out torque capability. It will need a bigger tilt cylinder to compensate and thereby a bigger hydraulic flow. Either a bigger pump is needed or a higher speed of the pump to supply the increased flow. The linkage is self-adjusting.

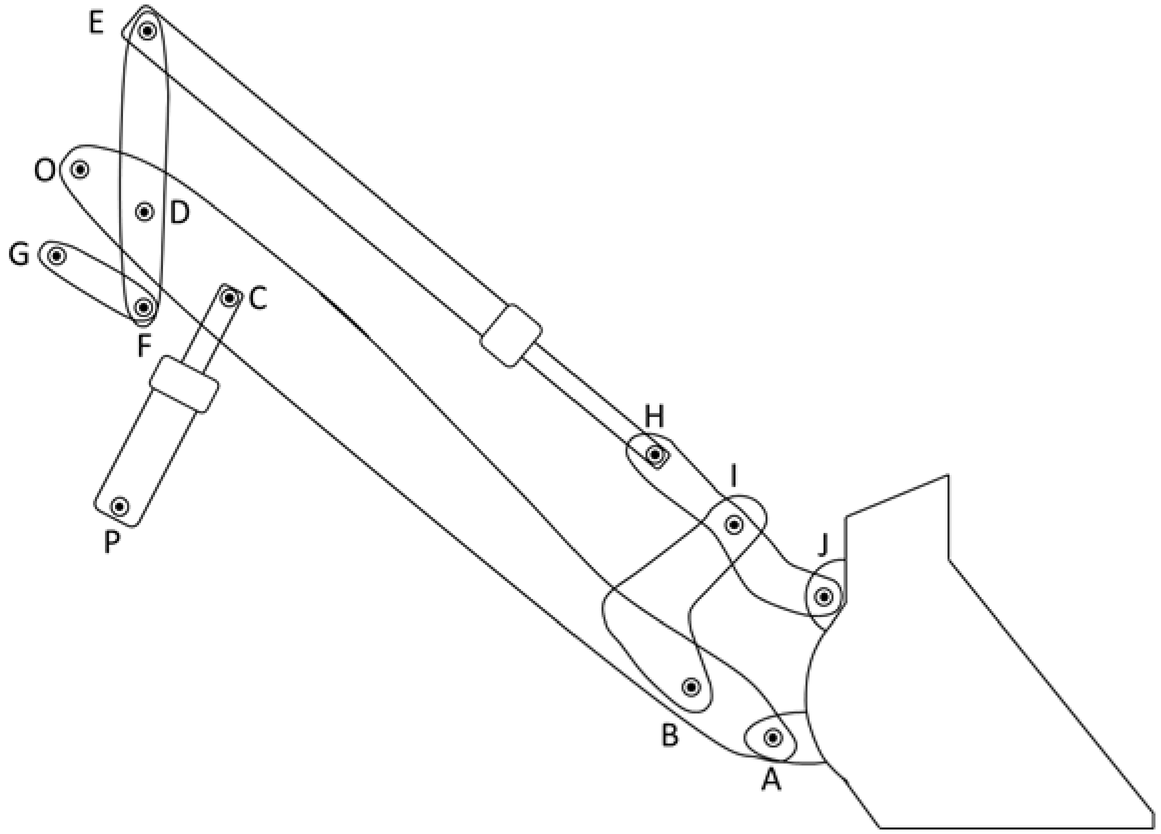

Figure 3.10 Example of a TBM-linkage.

3.2.4 TPC-linkage

The TPC linkage is shown in Figure 3.11. It is a stiff linkage i.e. it has no self-adjustment. Therefore it requires either constant control from the driver, an electro-hydraulic or mechanical-hydraulic compensation to keep it parallel. It is commonly used as front unit on backhoe loaders. It can have a good breakout torque for all positions. A problem could arise at the top position where the load actually can be dropped onto the drivers' cab since it is possible to roll back the bucket that far. On the other linkages introduced there is a natural mechanical stop that prevents it from dumping the load on the driver. It is a safety issue that needs to be dealt with in order to be able to use this linkage. On small backhoe loaders a mechanical feedback is often used to prevent the bucket from tilting back to much.

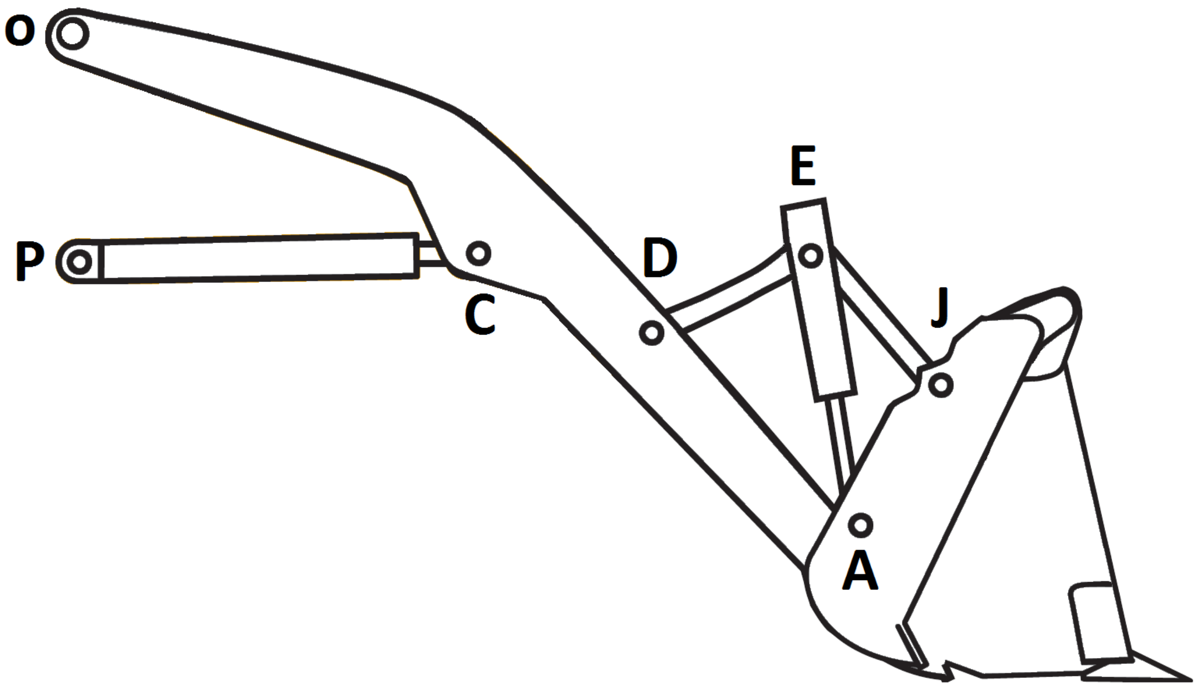

Figure 3.11 Example of a TPC-linkage.

3.3 Hydraulic system

The hydraulic system contains three pumps which are partly task specific and shown in Figure 3.12. One is for the working hydraulic system, one is for steering-, brake- and the working hydraulic systems (brake and steering prioritized) and one is for the brake- and cooling systems.

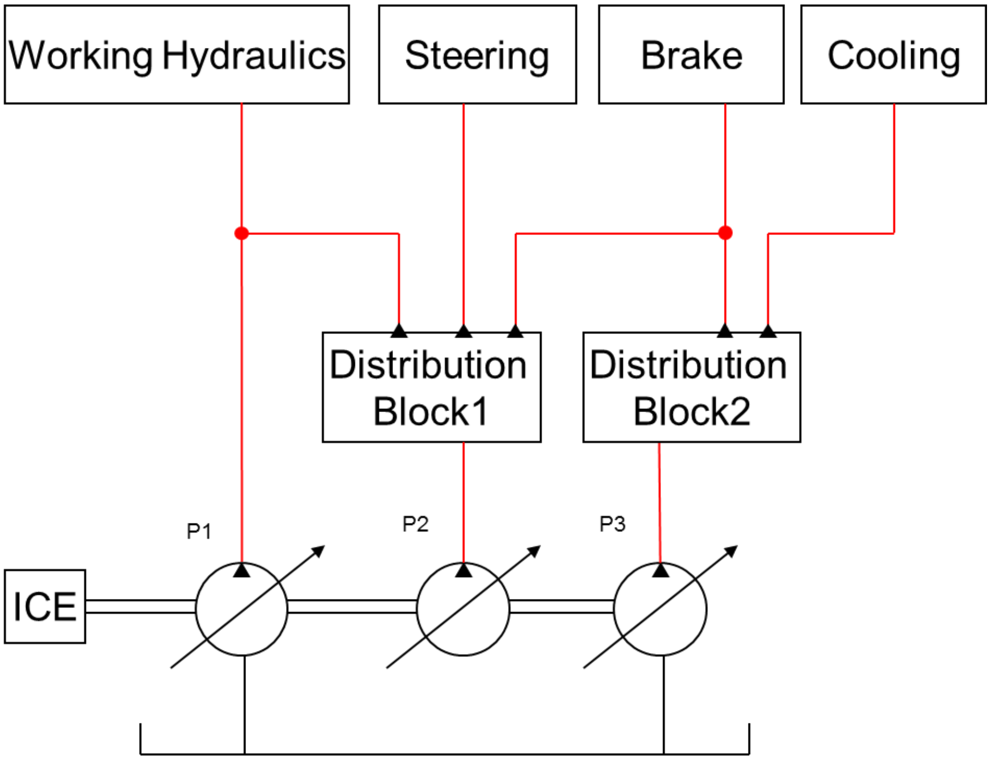

Figure 3.12 Simplified hydraulic system for the L120 series.

The pumps are load sensing, LS, with variable displacement. The WL L120s pumps are directly connected to the engine, which is controlled by the gas pedal.

The response time for the pumps is fast, about 0.1-0.2 second from zero to full displacement and about 0.05 from full to zero. The valves in the system have to be faster than the pump since the pump is supposed to follow the valves.

In the hydraulic system there is a refilling feature for the cylinders. This saves energy and prevents cavitation. It is based on the concept that the hydraulic fluid that leaves the cylinder is circulated back into the cylinder on the other side, decreasing the amount of fluid that the pump has to supply.

The working hydraulic system consists of many valves and will be described more thoroughly to get a better understanding of how a hydraulic system works and how complex it can be.

The lift cylinder is connected through a series of valves to a pump as in Figure 3.13 and represents the case when the cylinder is still. No flow will then run through the main valve.

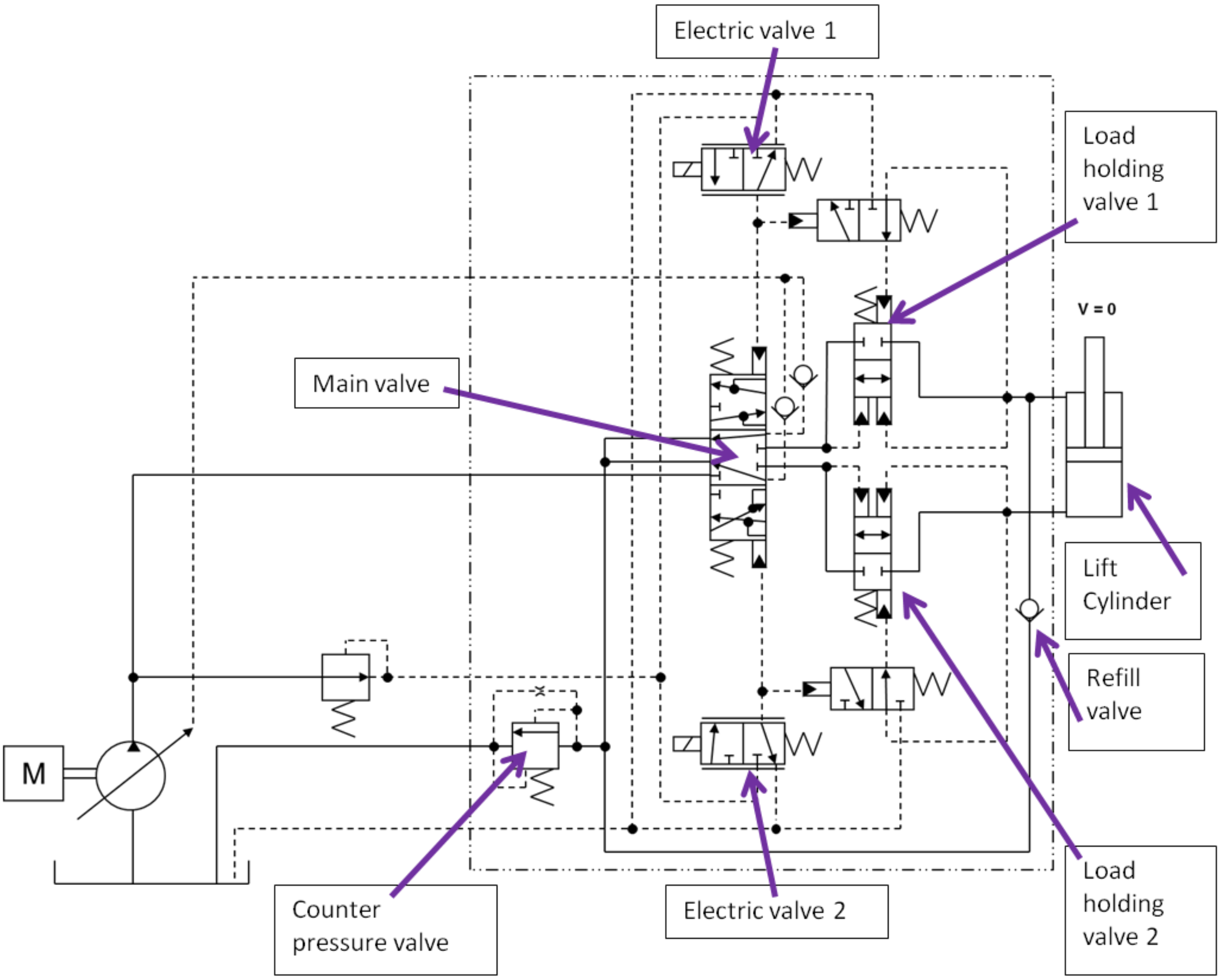

Figure 3.13 Hydraulic scheme for control of the lift cylinder, at rest [9].

To extend the lift cylinder the *electric valve 1* is activated as in Figure 3.14, look at Figure 3.13 for definitions. This action gives a pressure to the *main valve* and opens the flow to the cylinder. When the flow runs through, it also opens the *load holding valve 2* and runs to the plus side of the cylinder. From the minus side of the cylinder flow runs out to the *load holding valve 1* and through the *main valve*. Then it goes through the *counter pressure valve* and back to the tank. The colors of the lines in Figure 3.14 describes how the flow is lead. The red and green lines show the flow from pump while the yellow and blue shows the flow back to tank.

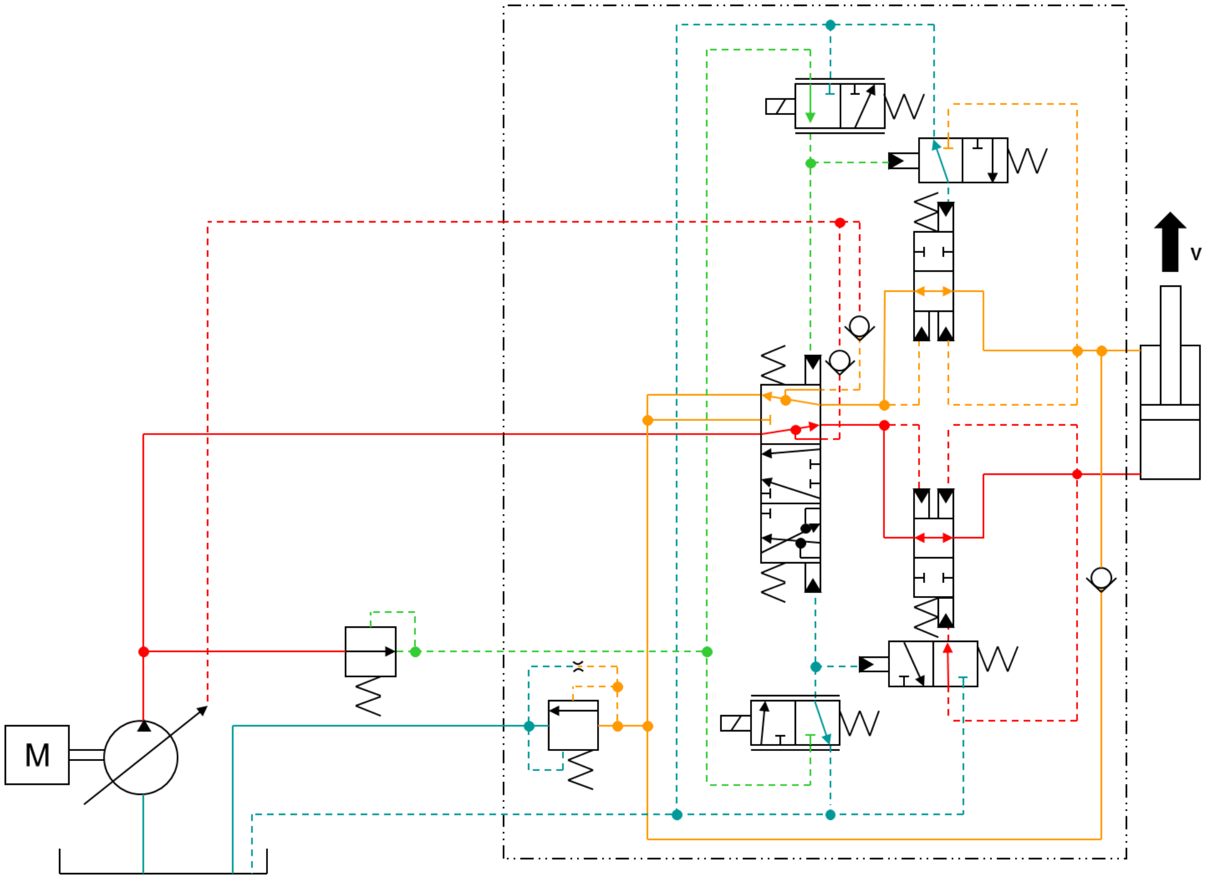

Figure 3.14 Hydraulic scheme for control of the lift cylinder, extending the cylinder [9].

To retract the lift cylinder the *electric valve 2* is activated as in Figure 3.15. This gives a pressure to the *main valve* and opens it. The flow then runs through the *main valve* and the *load holding valve 1* to the minus side of the lift cylinder. From the plus side of the lift cylinder the flow goes to the *load holding valve 2* and back to the *main valve*. After the *main valve* the flow could go either back to tank or back to the minus side of the cylinder, refilling it. It depends on if the pressure that is being built up by the *counter pressure valve* is greater than the pressure of the minus side of the lift cylinder. This saves a big amount of fluid that the pump no longer has to provide.

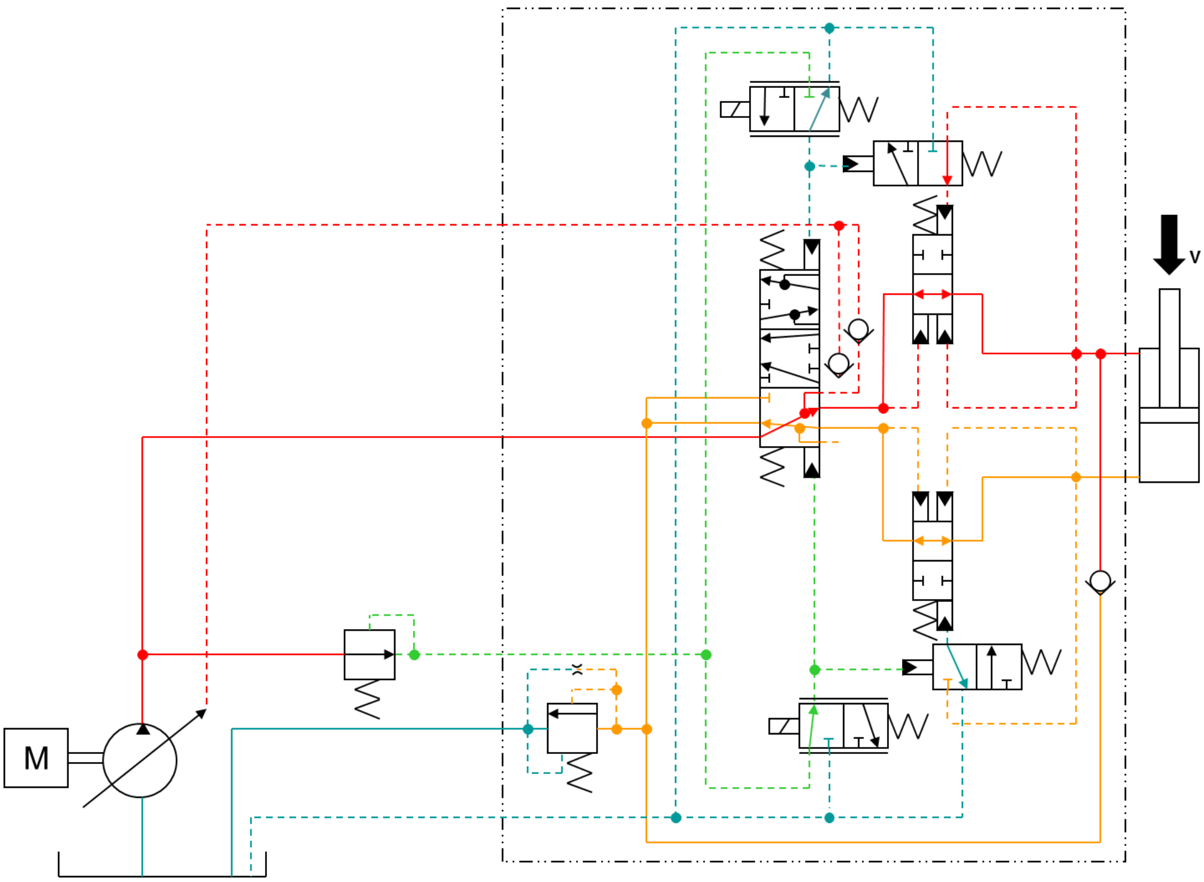

Figure 3.15 Hydraulic scheme for control of the lift cylinder, retracting the cylinder [9].

For the tilt cylinder the same concept is used except that it can refill the cylinder both when extending the cylinder and when retracting the cylinder. It is shown in Figure 3.16 that an additional check valve is added to the system, connected to the plus side of the tilt cylinder.

All the valves in Figure 3.13, Figure 3.16 and a few more, builds up the *working hydraulics* block in Figure 3.12.

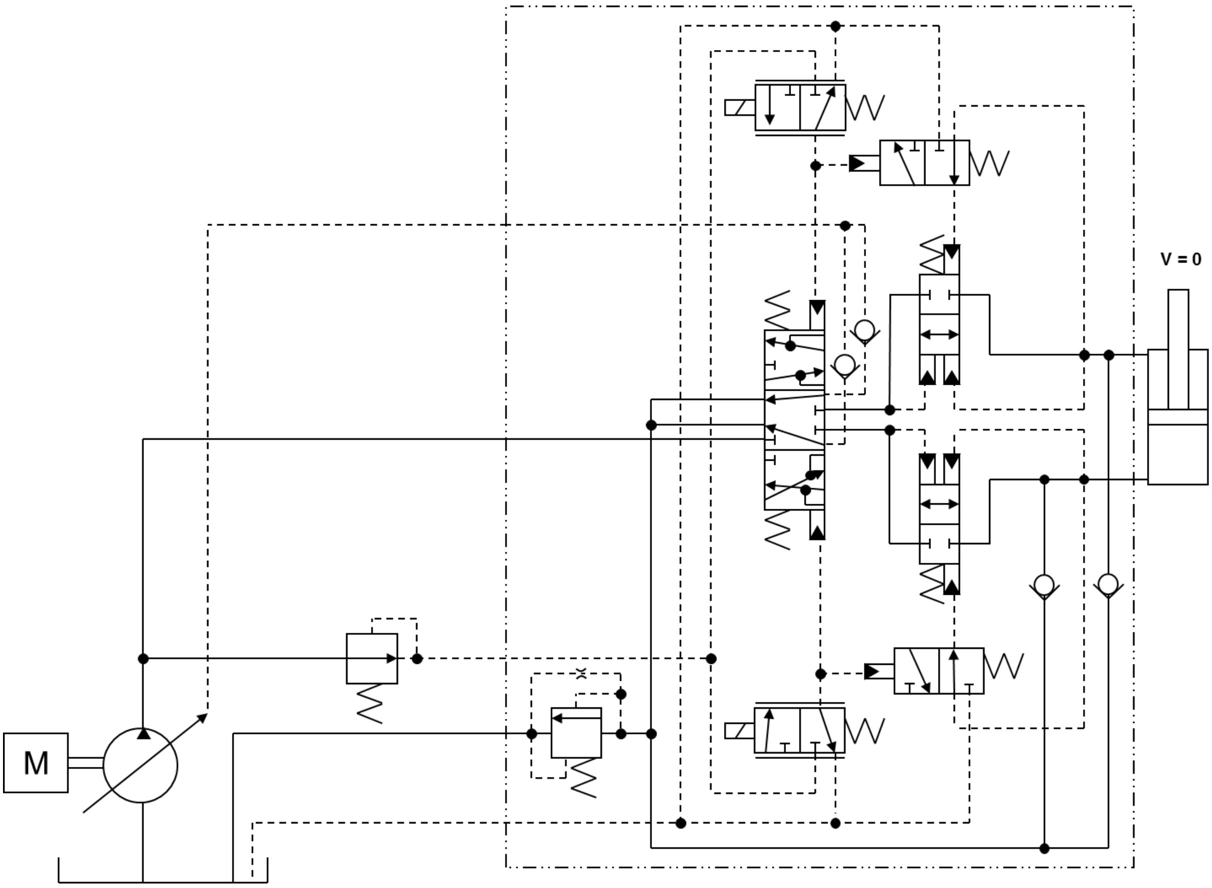

Figure 3.16 Hydraulic scheme for control of the tilt cylinder, at rest [9].

Chapter 4 – Development of a Work Cycle and Practical Limitations

There is a need of some limits of what is to be compared during the simulations. The first thing that needs to be specified is how the linkage is to be moved and under what conditions the hydraulic system should work. The models of the hydraulic and mechanical systems also have some simplifications and limitations.

4.1 Drive cycle

There are some time requirements when it comes to how fast the lift, emptying and lowering of a bucket should be done for a typical work load for the L120 series. It can be viewed in Table 4.1 for its max speed, at 2,100 RPM.

Table 4.1 –Time requirements L120 series at max RPM.

Cycle	Time [s]
Lift	5.4
Tilt	2.1
Lowering, empty	2.5
Total cycle time	10.0

A common drive cycle used for evaluation of WL is called “Short loading cycle” [9]; it can be viewed in Figure 4.1.

1. Bucket fill
2. Reverse from pile
3. Retardation to direction change
4. Direction change
5. Forward to load receiver
6. At the load receiver
7. Revers from load receiver
8. Retardation to direction change
9. Acceleration versus pile
10. Retardation to pile

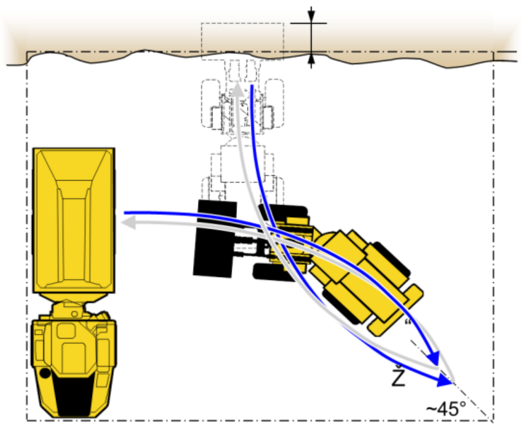

Figure 4.1 Short loading cycle.

Since this thesis work is concentrated on the energy consumption of different linkages a new drive cycle has to be specified without the part of the vehicle driving, a *Work Cycle*.

The work cycle will only lift, tilt and lower the bucket, based on the heights from Figure 3.3. First thought was that the work cycle will represent a WL that is standing still i.e. the engine at idle speed, but then it would not be possible for it to follow the time requirements of Table 4.1. A WL driver will although standing still actually press the gas pedal a bit to get extra power. Therefore the pump speed will be set to a fixed speed of 1,500 RPM during all simulations, about twice the idle speed. The time requirements are set at max RPM and since 1,500 RPM will be used instead, the work cycle will have to be longer than the time requirements specified in Table 4.1. The part of the work cycle when the linkage is being lowered should not take longer since the refill function should eliminate the need of pump work. A reference work cycle can be constructed through two different approaches.

1. The first approach specifies the lift height versus time and the tilt angle versus time, Figure 4.2 and Figure 4.3.
2. The second approach specifies the lift height versus time and the tilt angle as a function of the lift height. How the angle is dependent of the height can be viewed in Table 4.2 and Table 4.3 as well as in Figure 4.4 and Figure 4.5 for both the bucket and the fork tool.

Both approaches will be tried during this thesis work. The second approach is probably closer to how a WL is driven in reality. The references are based on data from three measured work cycles. The data displays the movement of the tilt and lift cylinder for a bucket and is shown in Appendix C.

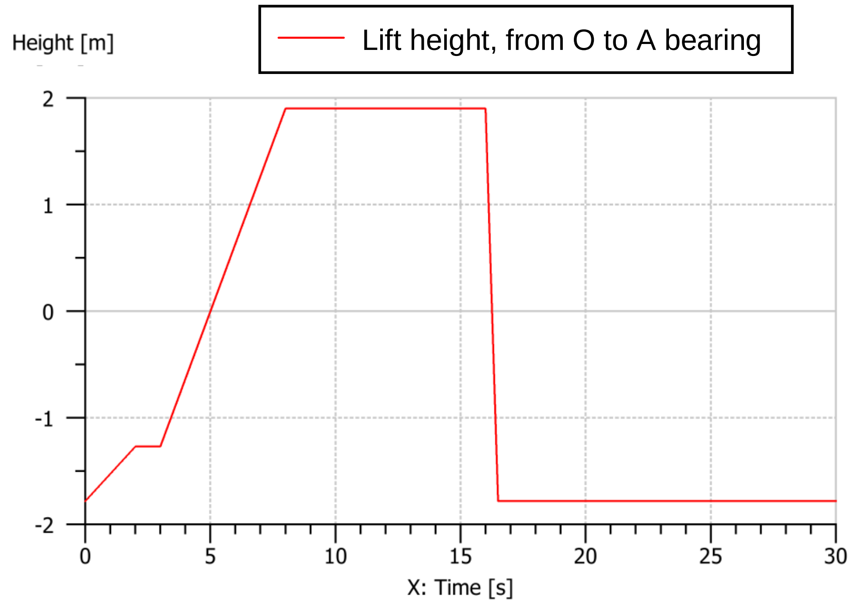

Figure 4.2 Height reference curve.

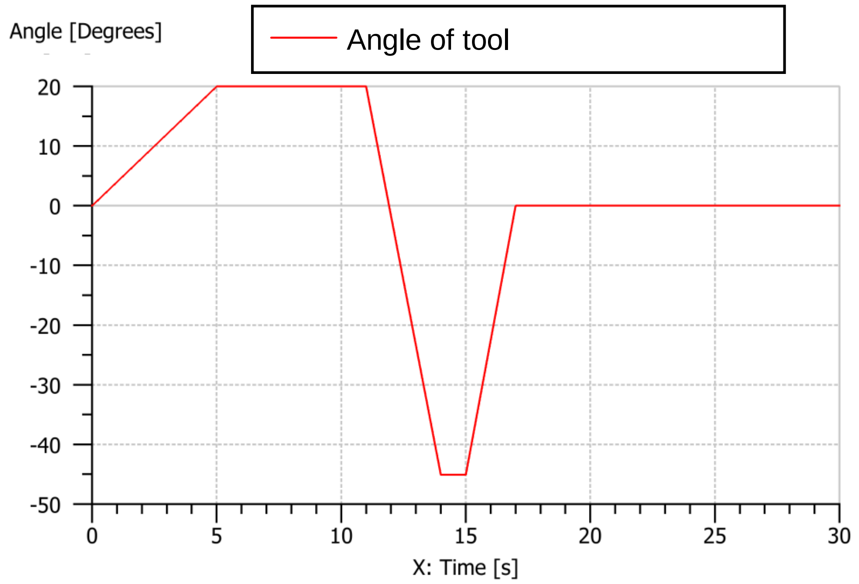

Figure 4.3 Angle reference curve for approach 1.

Table 4.2 Angle reference for bucket as a function of height.

Height of A-Bearing [m], Origin at O-bearing	Bucket-Cycle Angle [degrees], moving the linkage up	Angle [degrees], moving the linkage down
-1.7811	0	0
-1.3000	20	-10
-0.5000	20	-20
0	20	-45
0.5000	20	-45
1	20	-45
1.5000	20	-45
1.8000	20	-45
1.8500	20	-45
1.855	0	-45
1.8600	-45	-45

Table 4.3 Angle reference for forks as a function of height.

Height of A-Bearing [m], Origin at O-bearing	Fork-Cycle Angle [degrees], moving the linkage up	Angle [degrees], moving the linkage down
-1.7811	0	0
-1.3000	0	0
-0.5000	0	0
0	0	0
0.5000	0	0
1	0	0
1.5000	0	0
1.8000	0	0
1.8500	0	0
1.855	0	0
1.8600	0	0

The cycles will be run both with and without a load. The reason why, is that it is almost impossible to simulate the part of the work cycle when the bucket is being emptied. To simulate gravel leaving the bucket and the forces working on the bucket at that time demands very sophisticated software along with large CPU-power. Some linkages might be favored when no load is used and some might be favored when a load is applied. It is therefore of interest to do simulations both with and without a load so that all results can be analyzed.

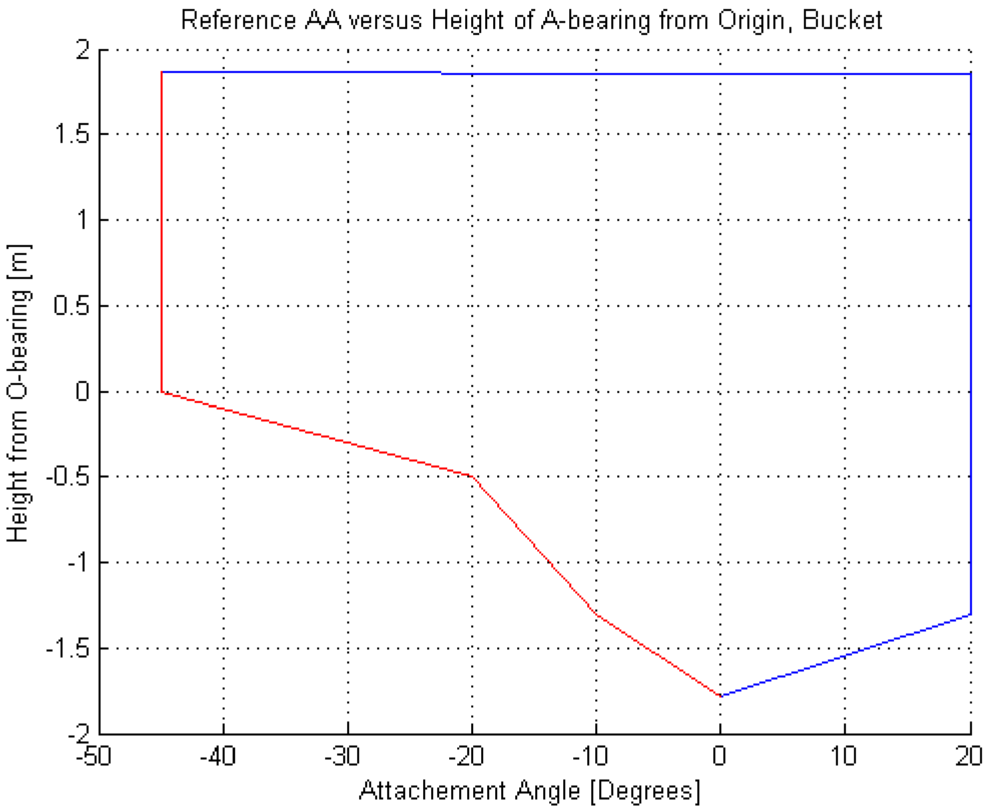

Figure 4.4 The blue curve shows the angle versus height of the tool when the linkage moves up and red curve shows the same thing on the way down.

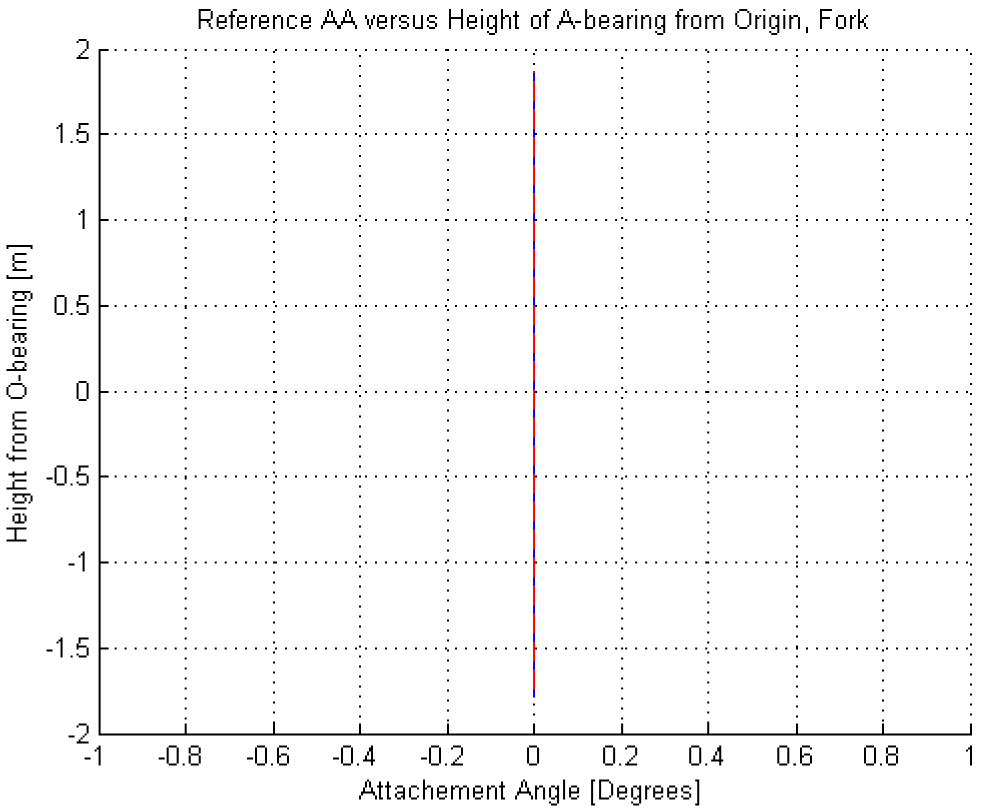

Figure 4.5 The blue curve shows the angle versus height of the tool when the linkage moves up and red curve shows the same thing on the way down.

4.2 Hydraulic system

The energy consumption will partly depend on how the hydraulic system behaves and therefore the same hydraulic system will be used for all linkages in order to make a fair comparison. The hydraulic system will be based on the one from L120 series, containing pumps, valves and regulators. However it is the functionality of the hydraulic system that is being modeled, not the actual one since it would have taken too much time to build an exact model. The model of the hydraulic system will be a simplification of the real one. The cylinders and some regulator parameters are linkage dependent.

Since the work cycle does not contain any driving i.e. no steering, both pumps connected to the working hydraulic system will be used to supply the loader linkage at all time. The third pump will not be used and therefore not modeled. The engine speed can also be looked upon as constant. Since both pump 1 and 2 will be used to supply the loader linkage it will be modeled as one pump with the total capacity of pump 1 and 2. The pump will use a mechanical efficiency map since that will have a big impact on the energy consumption.

The pumps and valves have certain response times that need to be fulfilled in the model. It cannot be too fast and not too slow. A step response analysis will be carried out on the system to ensure that the system is fast enough. The hydraulic system will have a refilling feature as described in Chapter 3.3.

4.3 Simulation

There are three cases that will be compared during the simulations of the bucket work cycle. For the fork work cycle only the first two cases below will be used.

1. In the first case the regulators only try to follow the lift reference curve in Figure 4.2, the tilt cylinder will not be regulated at all. This will be used as a reference curve for all other simulations (Unregulated Case).
2. In the second case the regulators will control both the tilt and lift at all time according to the reference curves and tables specified in chapter 4.1 (Electro-Hydraulic Compensated Case).
3. The third case will be a variant of the second one. Since a human does not drive as in any of the two cases above, a more human case is needed. A person will try to regulate the tilt in the beginning of the lift, when the bucket is being filled, and in the end, when the bucket is being emptied. The self-adjustment of the linkages releases the driver from using the tilt when the load is not sensitive i.e. a bucket of gravel (Human Case).

It can easily be understood that the third case is only of interest when using a bucket, not when driving a pallet since that type of load requires continuous compensation from the driver. The simulations will be conducted for both approaches mentioned in Chapter 4.1 and for all linkages.

4.4 Other limitations

No mass of the cylinders are taken into account, this because the center of mass will change when the cylinder stretches and rotates and is fairly hard to model. The mass could be set at a “middle” distance but that estimation is as good as not including the mass at all and is therefore left out. The mass of the cylinders are about 15 % of the total mass of the linkage. The friction in the joints are not taken into account for, they would probably give rise to a smother movement and higher energy consumption during the simulations. The reason that it is left out is that it would have been hard to integrate it in the model as well as finding data for what the friction would be.

Both the bucket- and the fork work cycle will use the same mass of the tool, i.e. the mass of the bucket will be used as the mass of the forks. This is done to be able to compare the energy consumption easier; it is possible to use the same reference curve. In reality the mass of the bucket is probably enough to cover both the mass of the forks and the load from a small pallet.

The fork work cycle will allow a small error of the tilt angle in a range of 5 degrees in total i.e. ±2.5 or +5-0 or equivalent. The reason for this is that 5 degrees is used as a limit for a WL that has this functionality on the market today [10].

Chapter 5 – Theory

The theory behind the simulations will be derived in the calculations that follows, both for the energy consumption and the total cost of ownership.

5.1 Energy calculations

The energy needed for a certain movement of a linkage can be calculated by analyzing the forces and torques that acts on the different links. The linkage can be looked upon as a two dimensional mechanism. Calculations for a Z-linkage are shown below and are derived from the definitions in Appendix B, where all forces and torques affecting the links are marked. For each linkage these calculations are different since they have different amount of links and joints. These calculations have to be done for every time increment and are the reason why simulations instead of calculations by hand are made. It is only shown for one linkage, the Z-linkage.

For link 1

$$(\uparrow): V_1 + V_2 + V_3 + V_4 - m_1 \cdot g = m_1 \cdot a_{1,y} \tag{5.1}$$

where $\boldsymbol{a_1}$ is the acceleration vector of the mass for link 1:

$$\boldsymbol{a}_1 = \ddot{\boldsymbol{r}}_1 = \boldsymbol{i}\ddot{x}_1 + \boldsymbol{j}\ddot{y}_1 + \boldsymbol{k}\ddot{z}_1 \tag{5.2}$$

and

$$a_{1,y} = \ddot{y}_1 \tag{5.3}$$

$$(\rightarrow): H_1 + H_2 + H_3 + H_4 = m_1 \cdot a_{1,x} = m_1 \cdot \ddot{x}_1 \tag{5.4}$$

The momentum around O CCW is:

$$M_1 - M_2 - M_3 - M_4 - m_1 \cdot g \cdot |OG_{1,x}| + V_2 \cdot |OA_x| + H_2 \cdot |OA_y| + V_3 \cdot |OC_x| + H_3 \cdot |OC_y| + V_4 \cdot |OD_x| + H_4 \cdot |OD_y| = m_1^2 \cdot (x_1 \cdot \ddot{y}_1 - \ddot{x}_1 \cdot y_1) \tag{5.5}$$

For link 2

$$(\uparrow): -V_4 - V_5 + V_6 - m_2 \cdot g = m_2 \cdot \ddot{y}_2 \tag{5.6}$$

$$(\rightarrow): -H_4 - H_5 + H_6 = m_2 \cdot \ddot{x}_2 \tag{5.7}$$

The momentum around G CCW is:

$$M_4 + M_5 - M_6 - m_2 \cdot g \cdot |GG_{2,x}| - V_4 \cdot |GD_x| + H_4 \cdot |GD_y| + V_6 \cdot |GF_x| + H_6 \cdot |GF_y| = m_2^2 \cdot (x_2 \cdot \ddot{y}_2 - \ddot{x}_2 \cdot y_2) \tag{5.8}$$

For link 3

$$(\uparrow): V_5 + V_7 - m_3 \cdot g = m_3 \cdot \ddot{y}_3 \quad 5.9$$

$$(\rightarrow): H_5 + H_7 = m_3 \cdot \ddot{x}_3 \quad 5.10$$

The momentum around G CCW is:

$$-M_5 - M_7 - m_3 \cdot g \cdot |GG_{3,x}| + V_7 \cdot |GJ_x| - H_7 \cdot |GJ_y| = m_3^2 \cdot (x_3 \cdot \ddot{y}_3 - \ddot{x}_3 \cdot y_3) \quad 5.11$$

For link 4

$$(\uparrow): -V_2 - V_7 - m_4 \cdot g = m_4 \cdot \ddot{y}_4 \quad 5.12$$

$$(\rightarrow): -H_2 - H_7 = m_4 \cdot \ddot{x}_4 \quad 5.13$$

The momentum around A CCW is:

$$M_2 + M_7 - m_4 \cdot g \cdot |AG_{4,x}| - V_7 \cdot |AJ_x| + H_7 \cdot |AJ_y| = m_4^2 \cdot (x_4 \cdot \ddot{y}_4 - \ddot{x}_4 \cdot y_4) \quad 5.14$$

Equations 5.1 to 5.14 can be put into an equation system as Equation 5.15:

$$\boldsymbol{AX} + \boldsymbol{B} = \boldsymbol{C} \quad 5.15$$

where:

$$\boldsymbol{X} = \begin{bmatrix} V_1 \\ V_2 \\ V_3 \\ V_4 \\ V_5 \\ V_6 \\ V_7 \\ H_1 \\ H_2 \\ H_3 \\ H_4 \\ H_5 \\ H_6 \\ H_7 \\ M_1 \\ M_2 \\ M_3 \\ M_4 \\ M_5 \\ M_6 \\ M_7 \end{bmatrix} \tag{5.16}$$

and:

$$\boldsymbol{A} = \left[\begin{array}{cccccccc:cccc}
1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
1 & 0 & 0 & 0 & 0 & 0 & -1 & 0 & |OA_x| & 0 & 0 & 0 \\
1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & |OC_x| & 0 & 0 & 0 \\
1 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & |OD_x| & -|GD_x| & 0 & 0 \\
0 & 0 & -1 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & 0 & |GF_x| & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 & -1 & 0 & 0 & 0 & |GJ_x| & -|AJ_x| \\
\hdashline
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 & 0 & 0 & 0 & -1 & |OA_y| & 0 & 0 & 0 \\
0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & |OC_y| & 0 & 0 & 0 \\
0 & 1 & 0 & -1 & 0 & 0 & 0 & 0 & |OD_y| & |GF_x| & 0 & 0 \\
0 & 0 & 0 & -1 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & -|GF_y| & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 & -1 & 0 & 0 & -|Gy| & |AJ_y| \\
\hdashline
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -1 & 0 & 0 & 1 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -1 & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & -1 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -1 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & -1 & 1
\end{array}\right]^T \tag{5.17}$$

and:

$$\boldsymbol{B} = \begin{bmatrix} -m_1 \cdot g \\ 0 \\ -m_2 \cdot g \\ 0 \\ -m_3 \cdot g \\ 0 \\ -m_4 \cdot g \\ 0 \\ -m_1 \cdot g \cdot |OG_{1,x}| \\ -m_2 \cdot g \cdot |GG_{2,x}| \\ -m_3 \cdot g \cdot |GG_{3,x}| \\ -m_4 \cdot g \cdot |AG_{4,x}| \end{bmatrix} \qquad 5.18$$

and:

$$\boldsymbol{C} = \begin{bmatrix} m_1 \cdot \ddot{y}_1 \\ m_1 \cdot \ddot{x}_1 \\ m_1^2 \cdot (x_1 \cdot \ddot{y}_1 - \ddot{x}_1 \cdot y_1) \\ m_2 \cdot \ddot{y}_2 \\ m_2 \cdot \ddot{x}_2 \\ m_2^2 \cdot (x_2 \cdot \ddot{y}_2 - \ddot{x}_2 \cdot y_2) \\ m_3 \cdot \ddot{y}_3 \\ m_3 \cdot \ddot{x}_3 \\ m_3^2 \cdot (x_3 \cdot \ddot{y}_3 - \ddot{x}_3 \cdot y_3) \\ m_4 \cdot \ddot{y}_4 \\ m_4 \cdot \ddot{x}_4 \\ m_4^2 \cdot (x_4 \cdot \ddot{y}_4 - \ddot{x}_4 \cdot y_4) \end{bmatrix} \qquad 5.19$$

Solving Equation 5.16 for $\boldsymbol{X}$ will give the cylinder forces as:

$$\boldsymbol{F_{c1}} = \boldsymbol{i}H_3 + \boldsymbol{j}V_3 \qquad 5.20$$

$$\boldsymbol{F_{c2}} = \boldsymbol{i}H_6 + \boldsymbol{j}V_6 \qquad 5.21$$

The force F_c that a cylinder can deliver is as Equation 2.1 states a function of the pressure and the area of the cylinder:

$$F_c = p \cdot A \qquad 5.22$$

The power, P, needed is stated in Equation 2.14:

$$P = q \cdot p \qquad 5.23$$

The energy, E, needed to move the linkage through a course can then be calculated by:

$$E = \int_0^t P\,dt \qquad 5.24$$

where P ia a function of the load and the position of the linkage. Expanding Equation 5.24 gives:

$$E = \int_0^t P\,dt = \int_0^t p \cdot q\,dt \qquad 5.25$$

were p is time dependent and q is dependent of the acceleration ($\frac{d}{t^2}$) of the linkage. This gives that the energy consumption is proportional to $\frac{1}{t^2}$. So the faster the movement is the more energy is needed. Therefore:

$$E \propto \frac{1}{t^2} \qquad 5.26$$

Equation 5.26 states that moving the linkage slowly will decrease the energy consumption radically. Therefore it is important that the simulations are carried out so that the linkages move at about the same speed, so the results will be comparable.

5.2 Cost calculations

Suppose two work cycles are possible every minute. Then the fuel consumption per hour can be calculated for the work cycle. It is fairly easy to understand that the fuel consumption will be different when running with a pair of forks than a bucket. The energy content [10] in one liter of diesel is:

$$Y_{Diesel} = 35.28 \cdot 10^{6} \tag{5.27}$$

The efficiency of a diesel engine is approximated to be constant although that is not the case for a real engine.

$$C_{Diesel} = 14{,}98 \tag{5.28}$$

$$\eta_{Diesel} = 0.35 \tag{5.29}$$

Together with the price of fuel [11] and the efficiency of a diesel engine, the cost of driving the WL with forks or bucket can be calculated per hour as:

$$C_{h,f} = \frac{E_f \cdot 2 \cdot 60}{\eta_{Diesel} \cdot Y_{Diesel}} \cdot C_{Diesel} \tag{5.30}$$

$$C_{h,b} = \frac{E_b \cdot 2 \cdot 60}{\eta_{Diesel} \cdot Y_{Diesel}} \cdot C_{Diesel} \tag{5.31}$$

Comparing the total cost for a linkage will have to take into account several things: how much time spent with a bucket versus forks, how many hours the machine is intended to run and the fixed cost of the linkage.

$$C_{manu} = m_l \cdot C_{mat} + m_l \cdot C_{proc} + m_c \cdot C_{cyl} \tag{5.32}$$

The total cost for a linkage per hour during its lifetime can be calculated through:

$$C_{Life} = \frac{S_f \cdot T_{Life} \cdot C_{h,f} + \left(1 - S_f\right) \cdot T_{Life} \cdot C_{h,b} + C_{manu}}{T_{Life}} \tag{5.33}$$

Chapter 6 – Modeling

Models are mainly built using the software *AMESim* for both the linkages and the hydraulic system. A regulator part is be built in *Simulink* and used to co-simulate with the part in *AMESim*. The models use the equations stated in the theory sections in Chapter 2 and Chapter 5.

6.1 Linkage model

Each linkage has a model of its own containing specific masses of the linkages and the geometries of the links and cylinders. A model of a linkage, as seen through the *AMESim* interface is shown in Figure 6.1. It might be hard to visualize this as a linkage and the corresponding physical layout is shown in Figure 6.2.

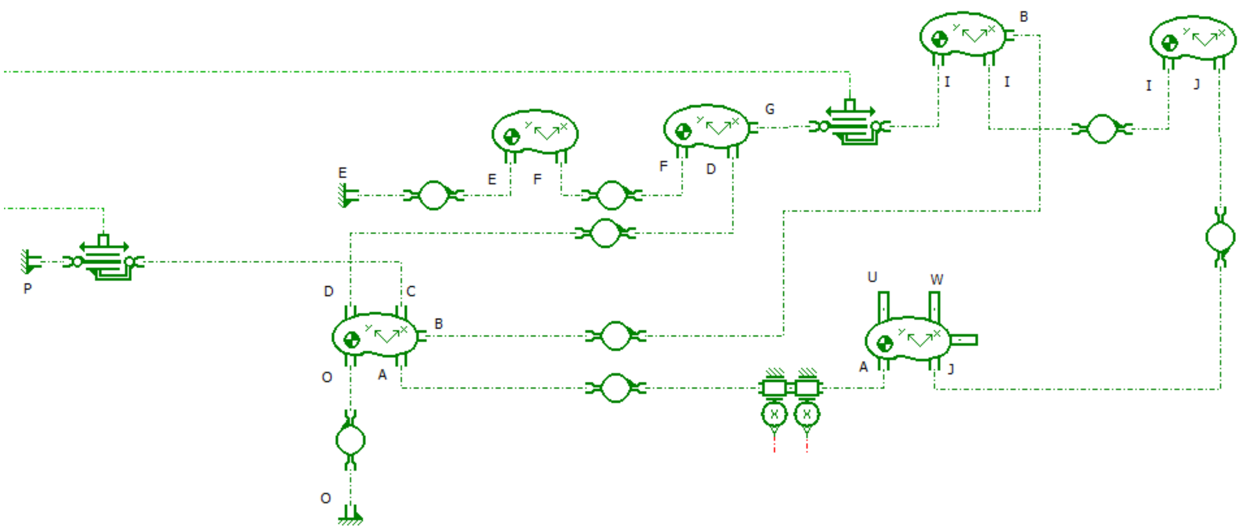

Figure 6.1 How a typical model of a linkage looks like in *AMESim*. This is the TBM-linkage.

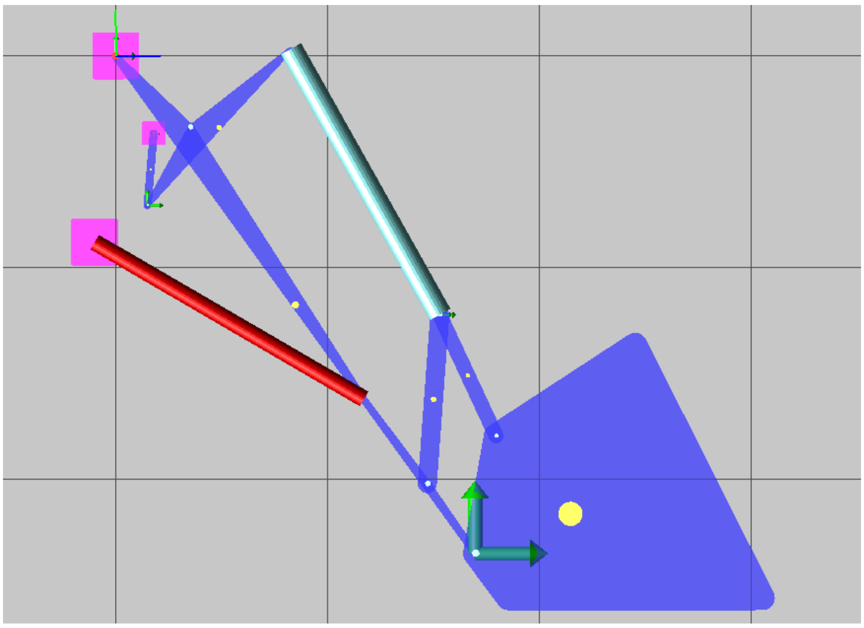

Figure 6.2 TBM-linkage.

6.2 Hydraulic model

The hydraulic system contains pump sizes and valves together with some regulators. In Figure 6.3 a hydraulic model is shown as it looks like in *AMESim*.

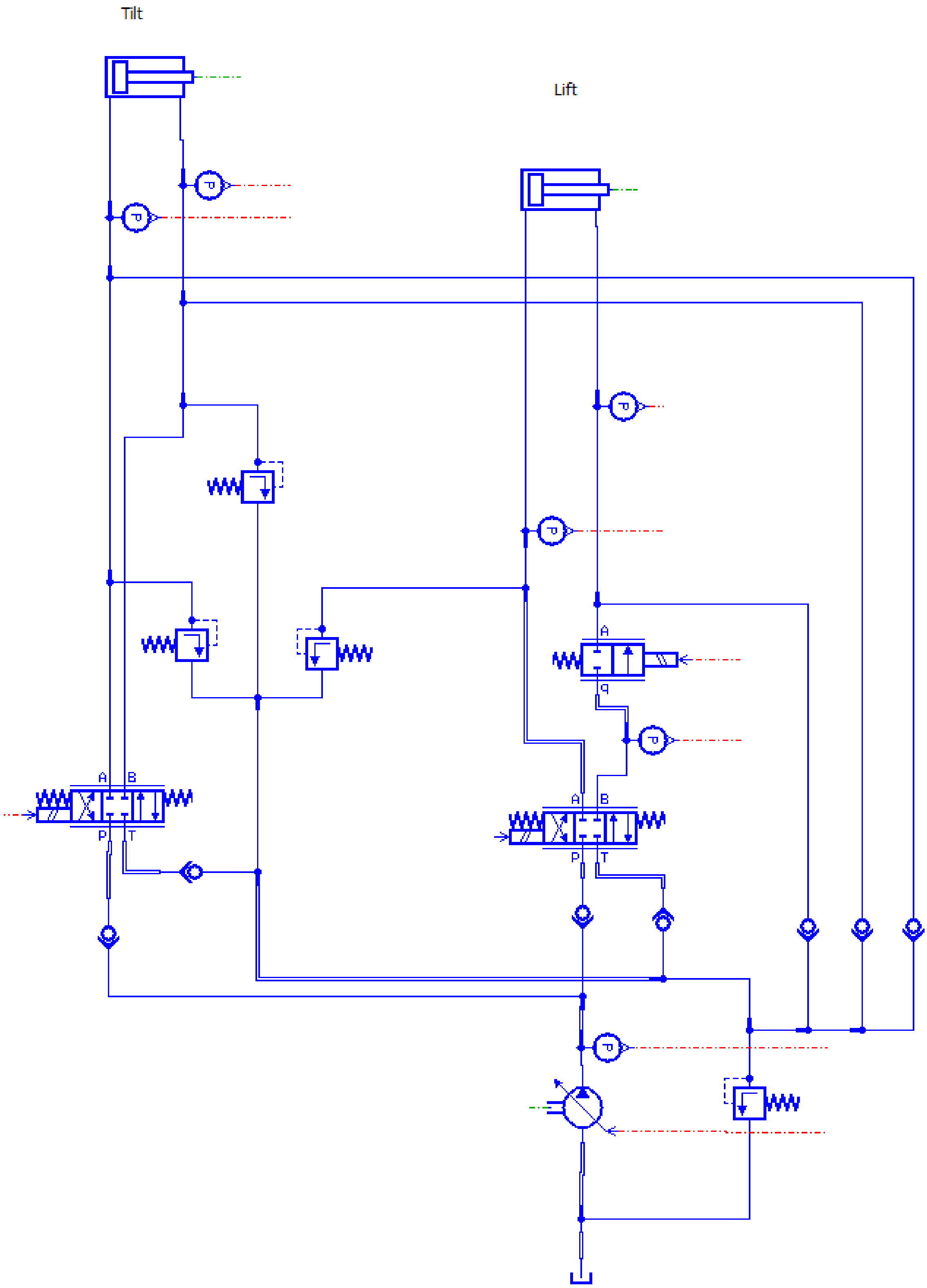

Figure 6.3 How a typical model of a hydraulic system look like in *AMESim*.

6.3 Control model

As explained before two approaches of control will be tried. The first one will have an angle reference and a height reference that is dependent on time, see Figure 4.2 and Figure 4.3. The second approach will have a height reference that is dependent on time and a reference angle that is dependent on the lift height, see Figure 4.2, Table 4.2 and Table 4.3. The different approaches require different control models and these can be viewed in Figure 6.4 and Figure 6.5. The regulators need to make sure that the system is as fast as the existing one on WL L120 series.

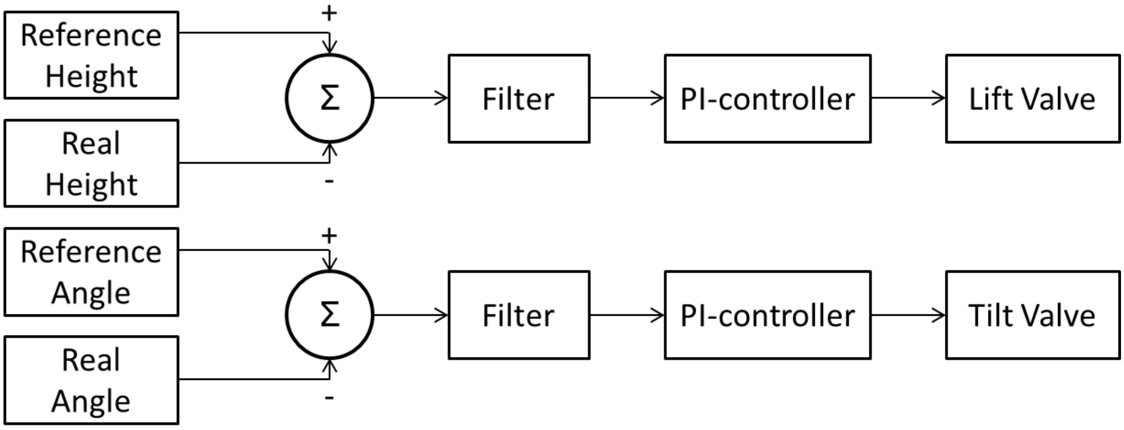

Figure 6.4 Control model used for approach 1.

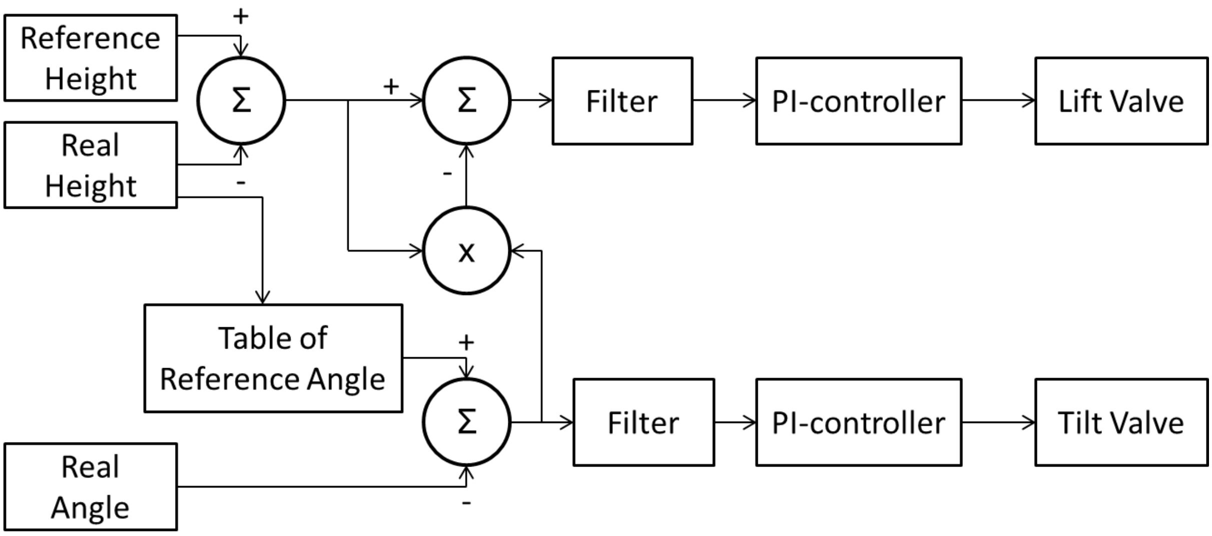

Figure 6.5 Control model used for approach 2.

6.4 Cost model

The cost calculations use flat figures for the efficiency of the ICE, material cost, processing cost and cylinder cost. The reason for this is that Volvo has not been able to provide their numbers for these calculations. The flat figures that are used are only estimates.

Chapter 7 – Results

Note, some data and results has been normed after the wish of Volvo CE.

7.1 Pump properties

To ensure that the hydraulic system is fast enough a step response of the pump displacement was made. A step change of the height reference point, from min to max, was made and the result can be viewed in Figure 7.1. First there is a delay time of about 0.08 seconds and then it takes about 0.12 seconds to increase the displacement from zero to maximum.

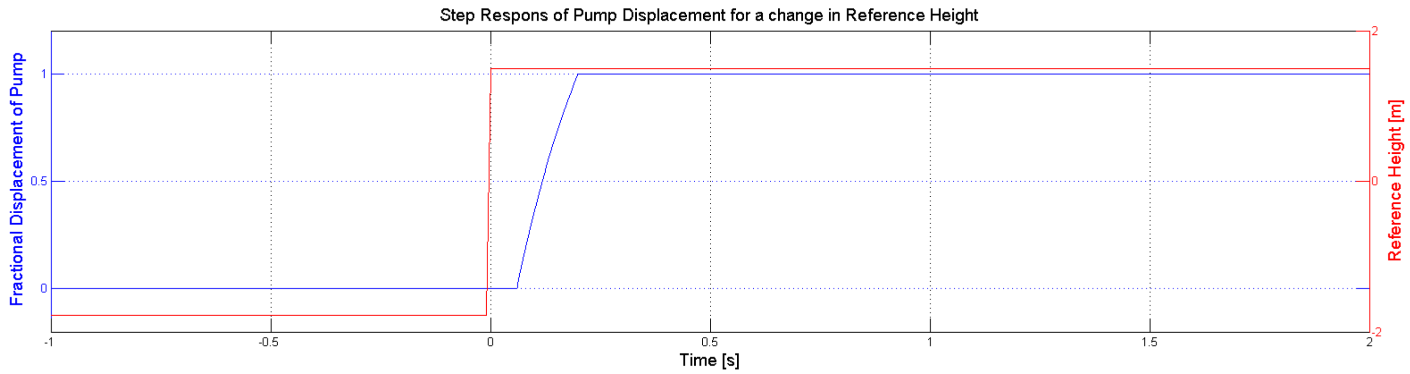

Figure 7.1 Step response of the pump.

7.2 Wheight

The masses of the linkages can be found in Table 7.1. As can be viewed the TP-linkage is the heaviest, Z-linkage is 4 % lighter. The TPC and TBM are 15 % and 13 % lighter than the TP-linkage respectively.

Table 7.1 Masses of the linkages including bucket and cylinders.

Linkage	Mass [%] of the TP linkage
TP-linkage	100 %
Z-linkage	96 %
TBM-linkage	87.6 %
TPC-linkage	85 %

7.3 Work cycles, operability and energy consumption

7.3.1 Work cycle approach 1

When the references for the lift height and the tilt angle were directly dependent of time the system did not always behave as expected. If the angle or height got too much behind their reference values, the system would end up locking itself. For instance when the tilt angle is impossible to reach at current lift height, then the pressure will be built up until maximum pressure is reached. When that happens a pressure release valve will open and the flow will be pumped through the release valve and back to tank. This result in no or just a small movement of the linkage. Sometimes the high pressures led to big oscillations that aborted the simulation. Because of all this the results from the simulations with approach 1 will not be presented. This model also had problems with implicit variables and following discontinuities while running the simulation.

7.3.2 Work cycle approach 2

For the second approach when the tilt angle was dependent on the lift height and the lift height was dependent on time, the refined control scheme solved any problem that could arise of locking the system up. The implicit variables where resolved in this approach by changing the setup of what solver type that was used for this approach. This solver did not have variables that were codependent. It is the results of this approach that is presented in this thesis.

The results from simulations are presented through diagrams and are found in Appendix D - Appendix H since there are so many. The results are commented in text below, the diagrams in the appendices can be looked into for a more extensive understanding of the simulation results.

7.3.3 Unregulated case without a load

The results from simulation of the unregulated case can be viewed as the green curves in Figure 7.2 for all the different linkages. Each row of diagrams presents the results for a specific linkage. The diagram to the left shows the total energy consumption, normed with the energy consumption of TP linkage, as function of time for a cycle. The middle diagram shows the height motion as a function of time (from O to A bearing) and the right diagram shows how the tilt angle of the linkage changes through time. The TP-linkage has the best self-adjustment followed by the Z-linkage and then the TBM and TPC linkages.

Another thing to notice is that all of the linkages follow the height reference curve well. The linkages differ when it comes to the time it takes to lower the linkage, TPC and Z are fastest followed by TP and last TBM. In reality it should take about 3 seconds to lower the linkage and TPC is the only one that can make it in that time. Z and TP make it in about 4 seconds, which is acceptable and TBM does it in 6 seconds. TBM is a bit too slow.

As can be seen Z has just a bit higher energy consumption than TP, TPC has really low energy consumption and TBM has the highest energy consumption. These will be used as reference curves when comparing the different cases of bucket and fork cycle simulations without a load, as stated in Chapter 4.3.

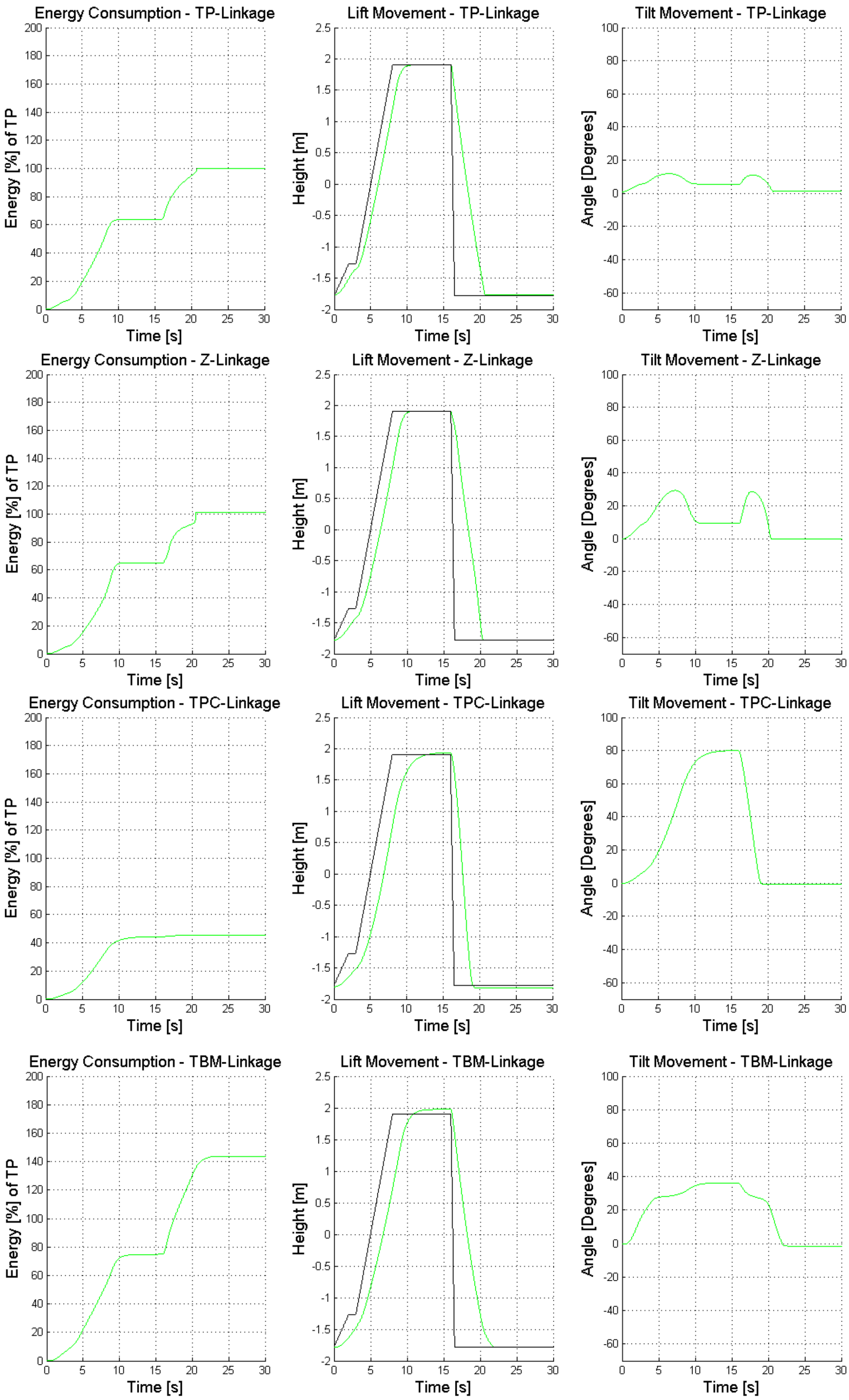

Figure 7.2 Results from the unregulated case. Each row of the diagrams presents the result for a specific linkage. The diagram to the left shows the energy consumption normed with the TP linkage, the middle shows the height of the linkage (from O to A bearing) and the right diagram shows the tilt angle of the linkage. The green curve is the movement of the linkage when only lifting it up and down. The black curve is the reference height.

7.3.4 Bucket work cycle without a load

For the bucket work cycle two simulations are made, the case when the regulators try to hold the correct angle and height all the time, *electro-hydraulic compensated case* (E-HCC) and the case when the regulators try to hold the correct height all the time and the correct angle for most of the time, the *human case*. The red curves in Appendix D present the *electro-hydraulic compensated case* and the blue curves present the *human case*. The green curves are the reference from previous figure, Figure 7.2, the *unregulated case*.

7.3.4.1 *Electro-hydraulic compensated case*

The blue curves viewed in Figure D.1 show that the linkages follow the height reference curve differently. TPC is fastest when it comes to lowering the bucket; TP is just a bit slower followed by TBM and last is Z that needs 9 seconds to lower the bucket. The Z-linkage has the lowest energy consumption followed by TPC, TP and last TBM.

The difference between a tilt cylinder that is overlying compared to one that is not overlying, can be viewed in Figure D.2, where the tilt position curve is inverted for the TBM linkage compared to the TPC linkage. Both the TP and the Z linkages need to expand and retract the cylinder compared to its original position but still reminds of the TPC linkages tilt movement.

The pressures for the linkages can be viewed in Figure D.3 - Figure D.6. There are a lot of oscillations for the TP and Z linkages, and some minor in the TPC linkage. There is a very high pressure in the plus side of the lift cylinder during the time 10-15 seconds for the TP and the Z linkages. The minus side of the tilt cylinder has at the same time very high pressure, for the same linkages. For TPC and TBM there is a lower pressure. The plus side of the tilt cylinder is high at time 15-20 seconds, during which the bucket rolls back at the same time as it is being lowered for TP and Z linkage. In the end of the cycle the minus side of the lift cylinder for TP and TBM maintain a high pressure.

7.3.4.2 *Human case*

The *human case*, the red curve, follows the height reference approximately the same way as the *electro-hydraulic compensated case* does. The tilt angle is almost the same too, except for TPC where it differs a lot. The reason for the different behavior of the TPC linkage is that it needs to be controlled all the time, since the tool will rotate the same as the boom around the O-bearing.

The Z-linkage has the lowest energy consumption followed by TP, TBM and last TPC.

The TPC is the only linkage that has higher energy consumption for the *human case* than for the *electro hydraulic compensated case*.

There are a lot of oscillations for the pressure in the TP linkage, and some minor in the TPC linkage. The cylinder pressures of the Z and TBM linkages do not have any oscillative behavior. The *human case* also shows the high pressures as described for the *electro hydraulic compensated case* between times 10-15 seconds for the TP and Z linkage. At the end of the cycle it also gets a maintained high pressure for the TP and TBM linkage in the minus side of the lift cylinder.

7.3.5 Fork work cycle without load

When driving with forks only the case of an *electro hydraulic compensation* of the parallel movement is of interest. A *human case* will be almost the same as the *electro hydraulic case* since a human needs to compensate manually at all time. The results can be viewed in Appendix E. The red curves present the *electro-hydraulic compensated case* and the green curves are the reference curves.

7.3.5.1 *Electro-hydraulic compensated case*

The parallel alignment is best for the TBM linkage shown in Figure E.1; it is almost parallel all the time. The other three linkages behave equally well and have a small error that stays within the five degrees margin. When it comes to following the height reference TPC is fastest at lowering the tool followed by TP, Z and last TBM.

The energy consumption for running this cycle is lowest for the TPC linkage, followed by TP, Z and last TBM.

The tilt cylinder for TP and Z linkages has to move more during the fork work cycle compared to the bucket work cycle. Its movement can be viewed in Figure E.2. Compared to Figure D.2 it is a very smooth motion for the tilt cylinder. The lift cylinder has the same motion as for the bucket work cycle.

The pressures for the linkages can be viewed in Figure E.3 - Figure E.6. There are only small oscillations when running the fork work cycle for the TP and the Z linkage. There is a very high pressure on the minus side of the lift cylinder at the end of the cycle for the TP and TBM linkages.

7.3.6 Unregulated case with 6 ton load

The *unregulated case* for a 6 ton load situation is shown in Figure 7.3. The TPC linkage has the lowest energy consumption followed by TP. Z linkages has slightly higher energy consumption than the TP linkage and TBM has the highest energy consumption.

The TPC is the fastest at lowering the forks followed by Z, TP and TBM. The green curve of the *unregulated case* will be used as a reference curve for the work cycles with a 6 ton load.

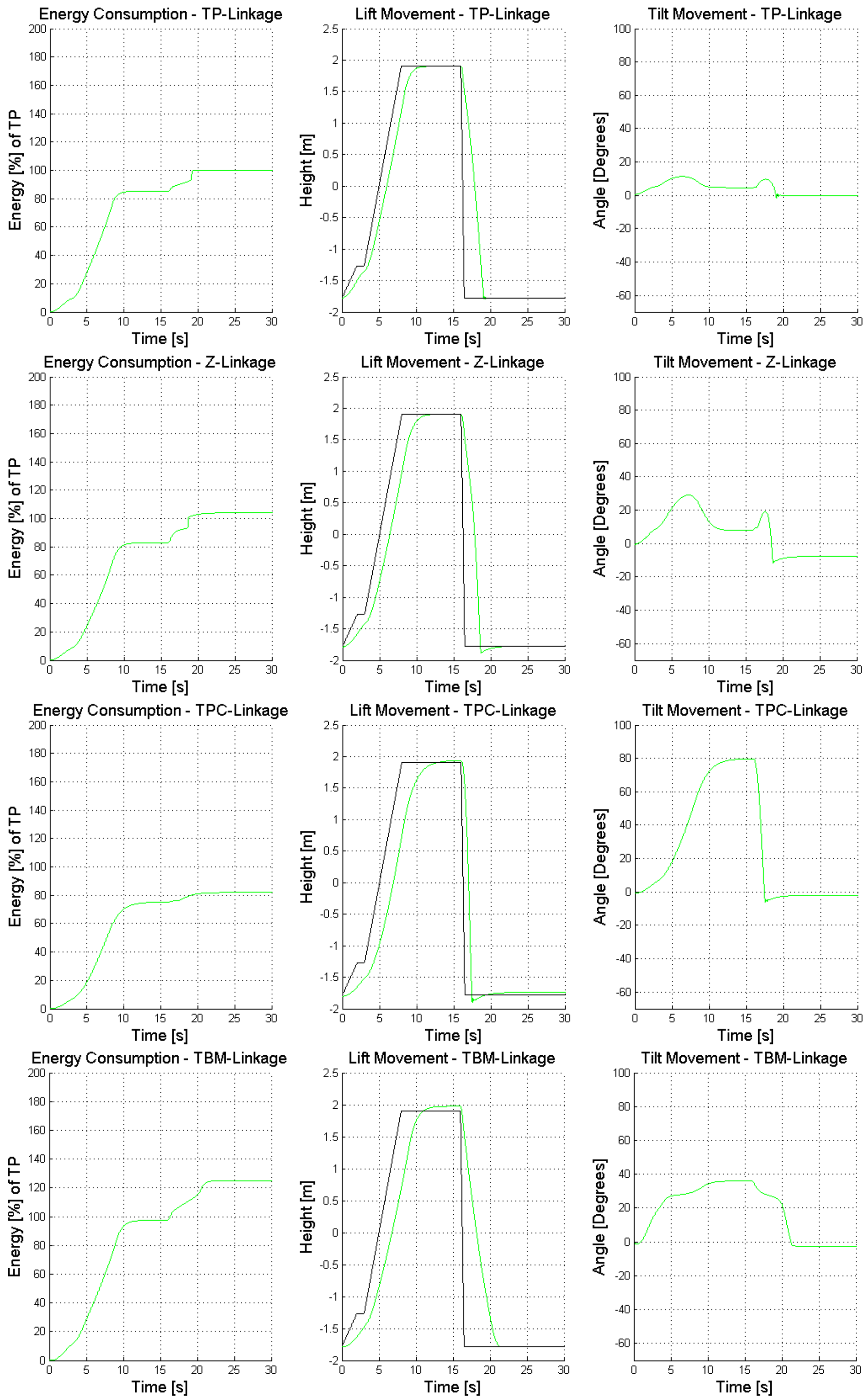

Figure 7.3 Results from the unregulated case with a 6 ton load. Each row of the diagrams presents the result for a specific linkage. The diagram to the left shows the energy consumption normed with the TP linkage, the middle shows the height of the linkage (from O to A bearing) and the right diagram shows the tilt angle of the linkage. The green curve is the movement of the linkage when only lifting it up and down. The black curve is the reference height.

7.3.7 Bucket work cycle with 6 ton load

The results from the bucket work cycle with a 6 ton load can be found in Appendix F. It is only of interest to look at the E-HCC for TP, Z and TBM and the human case for TPC since those are the more realistic work cycles. We can then see that TP is fastest at lowering the linkage followed by Z, TBM and last TPC. In this simulation the pressure in the cylinders oscillates a lot. The oscillations are so large that the linkage also starts to oscillate, since the pressure affects the length of the cylinders; it can be viewed in Figure F.1. The linkage that has the lowest energy consumption is TP followed by Z, TBM and last TPC. The oscillations takes place at the same time as in the work cycle when driving without a load.

7.3.8 Fork work cycle with 6 ton load

The results from running the fork work cycle with a load of 6 ton are shown in Appendix G.

7.3.8.1 *Electro-hydraulic compensated case*

As can be seen in Figure G.1 the TP linkage has the lowest energy consumption followed by Z, TBM and last TPC linkage.

The TP linkage is fastest at lowering the forks followed by TPC, Z and last TBM. The tilt angle of the TPC linkage fails the requirement of a maximum error of 5 degrees. It can also be viewed that the movement for all the linkages is somewhat oscillative, especially the tilt angle. It is possible to see those oscillations in the pressure curves of Figure G.3 - Figure G.6 as well as in the cylinder displacements in Figure G.2.

The results are summarized in Table 7.2 and Table 7.3.

Table 7.2 Summary of energy consumption for the different linkages.

	Summary of energy consumption (Bucket work cycle uses: E-HCC for TP,Z, and TBM, *human case* for TPC).			
Linkage	**Bucket work cycle without a load**	**Fork work cycle without a load**	**Bucket work cycle with a 6 ton load**	**Fork work cycle with a 6 ton load**
TP	2	2	1	1
Z	1	3	2	2
TPC	3	1	4	4
TBM	4	4	3	3

Table 7.3 Summary of lowering speed for the different linkages.

	Summary of lowering speed (Bucket work cycle uses: E-HCC for TP,Z, and TBM, *human case* for TPC).			
Linkage	**Bucket work cycle without a load**	**Fork work cycle without a load**	**Bucket work cycle with a 6 ton load**	**Fork work cycle with a 6 ton load**
TP	2	2	1	1
Z	4	3	2	3
TPC	1	1	4	2
TBM	3	4	3	4

7.4 Improved work cycle

Instead of calculating the cost for all the work cycle simulations, both with and without a load and compare them to each other, an improved work cycle is introduced.

When using a WL with a bucket it is most common to lift gravel from the ground up to a high position, dump the load and then lower the bucket while empty. This kind of work cycle was not possible to simulate. But what is possible is to combine the bucket work cycle with and without a load in order to get a work cycle that mimics the desired work cycle. By adding one part from the work cycle with a load together with one part from the work cycle without a load a more realistic work cycle can be constructed. The same thing can be done for the results from the fork work cycle. For the fork work cycle the E-HCC is used. For the bucket work cycle the *human case* is used for all linkages except TPC, since that does not represent a valid work cycle. TPC instead uses the E-HCC.

This is illustrated for the fork work cycle of the TP linkage. The energy used between t=0-16, Figure 7.4 (red curve), with a load plus the energy used between t=16-30, Figure 7.5 (red curve), without a load give a good approximation of the energy needed for the desired, more realistic, work cycle. This energy consumption is used as E_f in Equation 5.30. For the bucket work cycle the same thing is also done and is used as E_b in Equation 5.31.

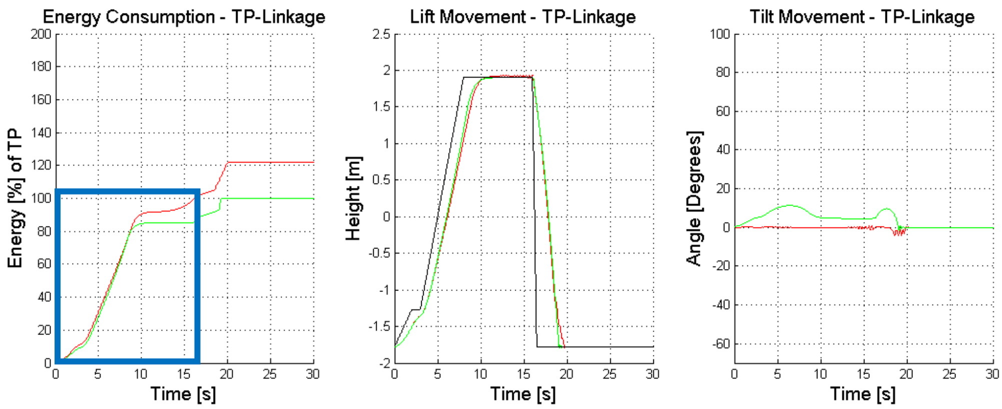

Figure 7.4 TP linkage, fork work cycle with 6 ton load.

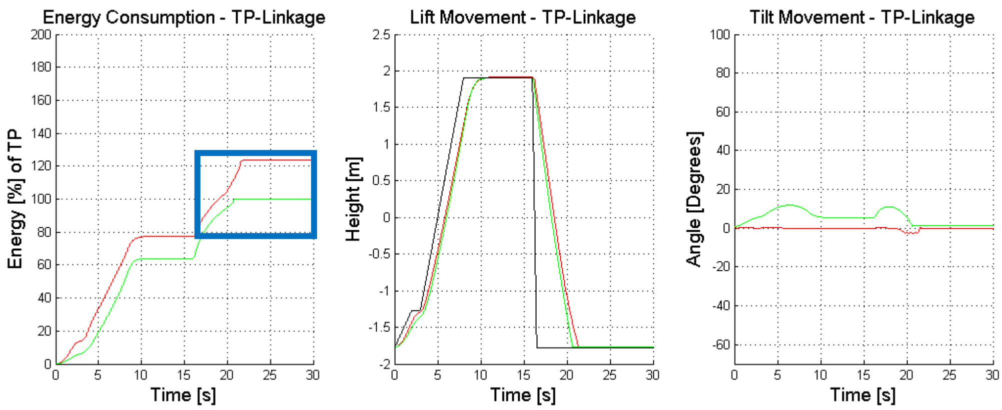

Figure 7.5 TP linkage, fork work cycle without load.

7.5 Cost

Since Volvo CE requests to keep the results anonymous of which linkage is which from this point on, that will be done. The linkages will be adressed as; ***Linkage 1*** in Black, ***Linkage 2*** in blue, ***Linkage 3*** in red and ***Linkage 4*** in green in the diagrams.

Table 7.4 Color description of linkages.

Linkage No.	Color
Linkage 1	Black
Linkage 2	Blue
Linkage 3	Red
Linkage 4	Green

The results from the cost calculations of Equation 5.33 for $0 < S_f \leq 1$ and for $1{,}000 \leq T_{Life} \leq 20{,}000$ are shown in Figure 7.6. The stars in Figure 7.6 make up a whole surface for the cost as a function of S_f and T_{Life}. By looking at the 3-D plot from underneath the view of Figure 7.7 appears. It is easy to see that ***Linkage 2*** is the cheapest linkage almost throughout the whole spectrum except for when only driving with forks $S_f \rightarrow 1$ and $T_{Life} \rightarrow 0$. If the vehicle is only driving with forks ***Linkage 3*** is better. The 3-D plot of Figure 7.6 can be considered from one side at a time and is displayed in Appendix H.

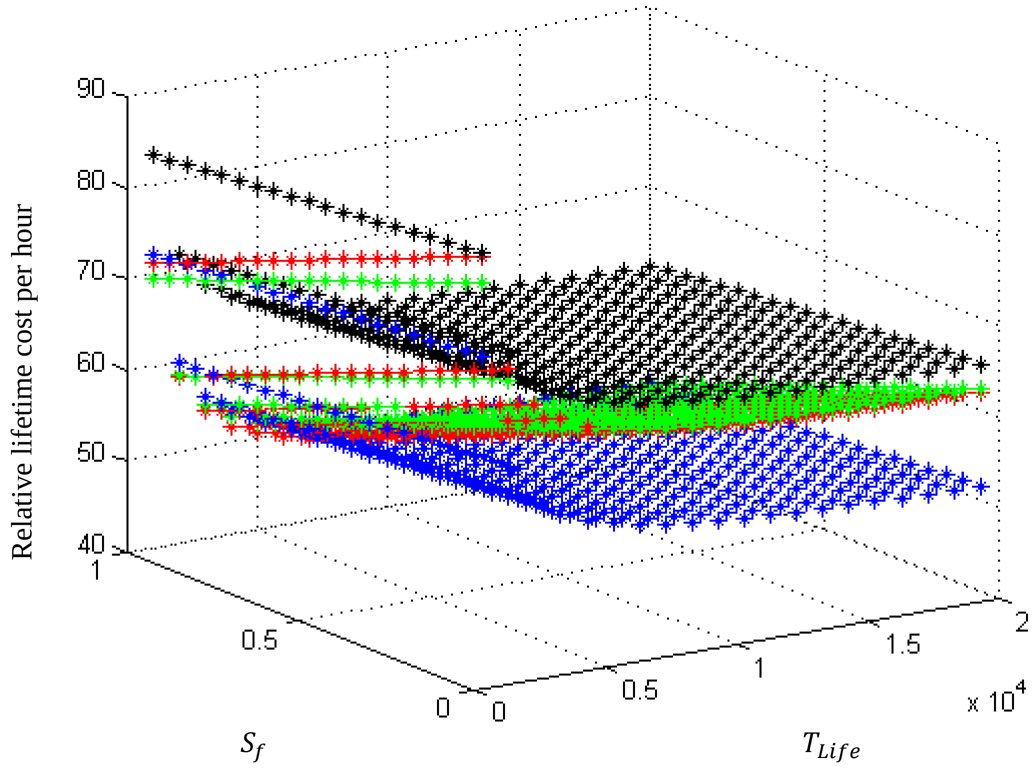

Figure 7.6 Map of relative lifetime cost per hour for the linkages as a function of S_f and T_{Life}.

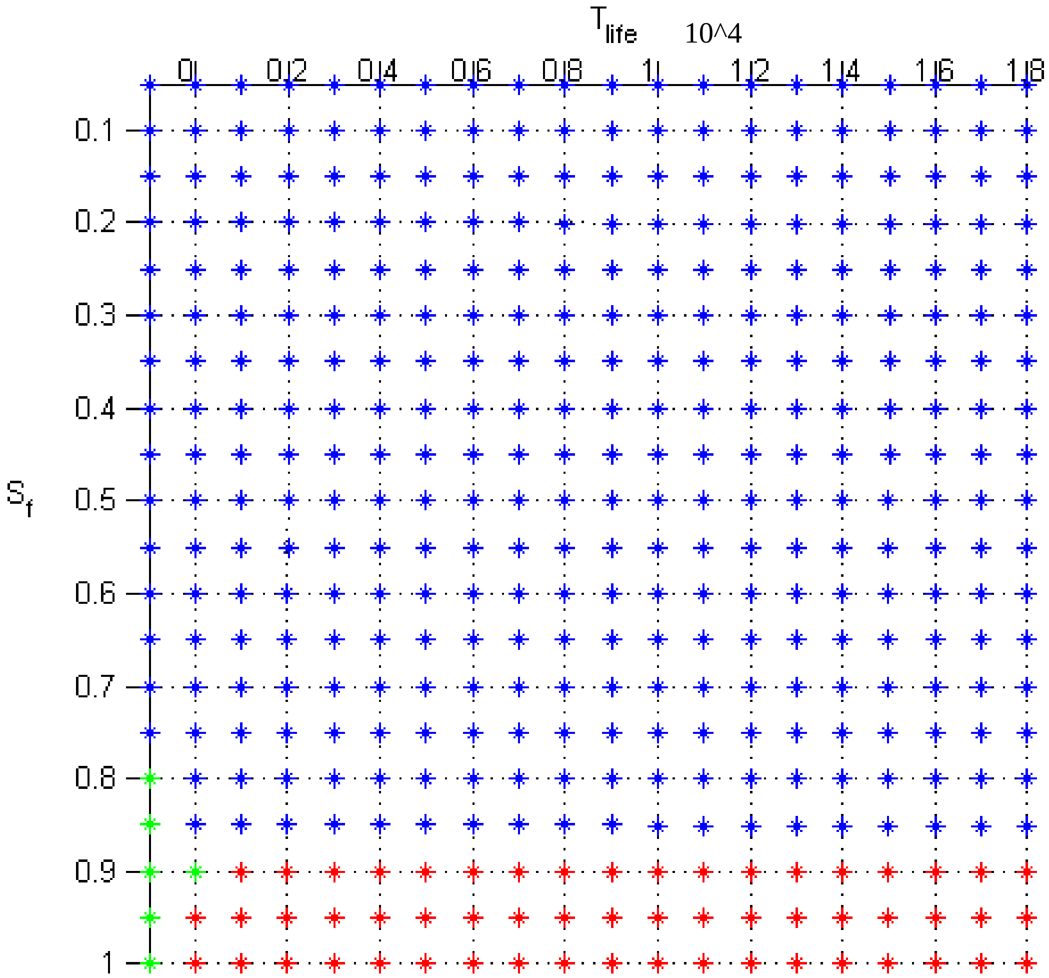

Figure 7.7 Lowest relative lifetime cost per hour for the linkages as a function of S_f and T_{Life}.

When the vehicle only is used for a few work hours (1,000 as in Figure H.1) and only using forks to lift pallets all the time, then ***Linkage 4*** is cheapest followed by ***Linkage 3***, ***Linkage 2*** and ***Linkage 1***. Another order is presented when only a bucket is used, then ***Linkage 2*** is cheapest, followed by ***Linkage 4, Linkage 3*** and ***Linkage 1.*** When the vehicle is used for many hours (20,000 as in Figure H.2) and only forks is being used the cheapest is ***Linkage 3*** followed by ***Linkage 4, Linkage 2*** and ***Linkage 1***.

Table 7.5 Summary of life time cost for the linkages.

	Summary of life time cost, position in order from lowest to highest.			
Linkage	$T_{Life} = 1,000$ $S_f \to 0$	$T_{Life} = 1,000$ $S_f \to 1$	$T_{Life} = 20,000$ $S_f \to 0$	$T_{Life} = 20,000$ $S_f \to 1$
Linkage1	4	4	4	4
Linkage2	1	3	1	3
Linkage3	3	2	2	1
Linkage4	2	1	3	2

Chapter 8 – Discussion

8.1 Pump properties

The results for how the pump behaves in the simulated system fits well with the real response times of the pumps as stated in Chapter 3.3. The pump response time indicates that the model of the hydraulic system is fast enough.

8.2 Wheight and operability

The TP linkage has the best self-adjustment followed by Z, TBM and last TPC, which cannot be driven as it is. All the linkages can successfully be used parallel when lifting the linkage up and down with the help of an electro-hydraulic compensation. The TPC linkage had a small error in its parallel movement with a full load and running with forks but that depends on the regulators, not the linkage itself.

The author think it is interesting that even though there is a big difference when it comes to the mass of the linkages, the energy consumption of the linkages does not correlate to the heaviest linkage. For instance, the TBM linkage is the second lightest but has the highest energy consumption for both work cycles with and without load; compare Table 7.1 with Figure D.1 and Figure E.1. This will clash with the reasoning that a lower weight of the linkage will decrease the amount of work that has to be done for every lift. A smaller linkage would also reduce the counter weight of the vehicle.

8.3 Work cycles and energy consumption

The first approach on how to define the work cycle did not turn out very good. The simulation program could not handle the implicit variables and discontinuities. Sometimes the simulation would not run through or the simulation did not run as expected. As a result no data could be extracted. Therefore more time were spent to develop the second approach and design it more carefully.

The second approach worked much better and is more advanced. By a change of setup, no implicit variables got created, solving the problem with implicit variables. The new control setup starts to close the lift cylinder valve whenever the tilt angle has a bigger error than 2 degrees. This way it is guaranteed that the linkage does not end up in a position it is not meant to be in. That removes the risk of the linkage getting stuck or move in a behavior that is undesirably. I think that the result when using this approach is more interesting since it is closer to how a real driver would use the linkage. As stated before it is only the result of the second approach that is presented in this report.

8.3.1 Unregulated case without load

When lowering a linkage all hydraulic fluid needed should go from the plus side to the minus side of the lift cylinder, except for the TBM linkage that need some extra fluid from the pump. This has not been the case during the simulations. The reason that the linkages move slowly, TBM in particular, is probably associated with the simplifications made when designing the model of the hydraulic system.

Only one linkage has fallen freely and the pump has had to work in order to push the other linkages down. TP, Z and TBM have had this problem. TPC is the only linkage that has fallen freely. This is shown in Figure 7.2 where there is an increase of total energy used after time t=16. If the linkage falls freely there will not be an increase of energy consumption, as it is for all linkages during simulations except TPC. The circulation of hydraulic fluid between plus and minus side is slower in the model than in reality.

It is important to remember that the TBM linkage actually needs supply from a pump to avoid cavitation when the linkage is being lowered since it has an overlying tilt cylinder. Since no free fall happens, the results for the energy consumption have to be looked upon with care, though it is still believed that the tendency of what linkage that uses most energy is correct since the linkages moves as desired. If only the part from start until the linkage is being lowered (0-16 seconds) of the work cycle is considered it states that the TPC linkage has the lowest energy consumption for cycles without a load. Followed by TP and Z that are about equal and last TBM.

8.3.2 Bucket work cycle without a load

The reason why the Z-linkage has the lowest energy consumption for both the fork and the bucket work cycle without a load is probably because it is slower than the other linkages. A consequence of Equation 5.26 is that the energy consumption is proportional to $\frac{1}{t^2}$. So the longer time it takes to lower the linkage the less energy is needed. It also gives more time for the re-circulation process between the plus and minus side of the cylinder, which would decrease the need of the pump to work and thereby reduce the energy consumption more.

The reason why TPC has higher energy consumption for the *human Case* than the *electro-hydraulic compensated* case is because of its geometry. It is not possible to use this linkage without constant control, which can be viewed in Figure 8.1 for the *human case* (blue curve). Therefor the *human case* for the TPC linkage should not be considered. Although it can be used to state the fact that the linkage is unsuitable to use without continuously compensation. The tilt angle varies a lot for TPC linkage for the unregulated case, up to 80 degrees. That is four times more than what the TP linkage does. That motion does not follow the recommendation of [5] and probably costs a lot of energy.

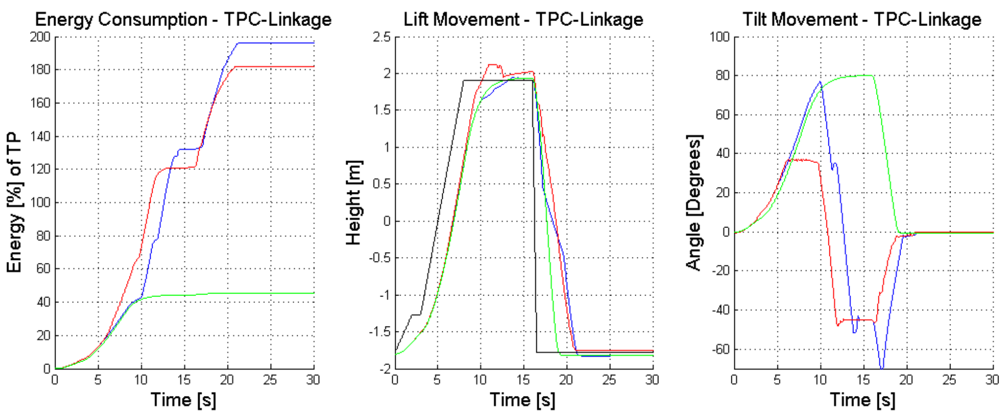

Figure 8.1 Energy consumption for TPC linkage for bucket work cycle without a load. The green curves are the *reference case*, the blue is the *human case*, and the red is the *electro-hydraulic compensated case*.

It is expected that TBM has somewhat higher energy consumption since its tilt cylinder is over lying. The results also showed that it had the highest total energy consumption for the bucket work cycle (when the *human case* for TPC was disregarded) and for the fork work cycle as well. This design will require a bigger flow for the same amount of work from the cylinder when tilting in and that is the reason to the high total energy consumption.

The high pressures at the end of the work cycle in the minus side of the lift cylinders for TP and TBM linkages suggest that a pressure is locked into the cylinder as in Figure 8.3. It can happen on a real WL and does not seem to affect the energy consumption. The energy consumption is zero after time t=23 seconds in Figure 8.2 (yellow), when the linkage has stopped moving, but the high pressure remains.

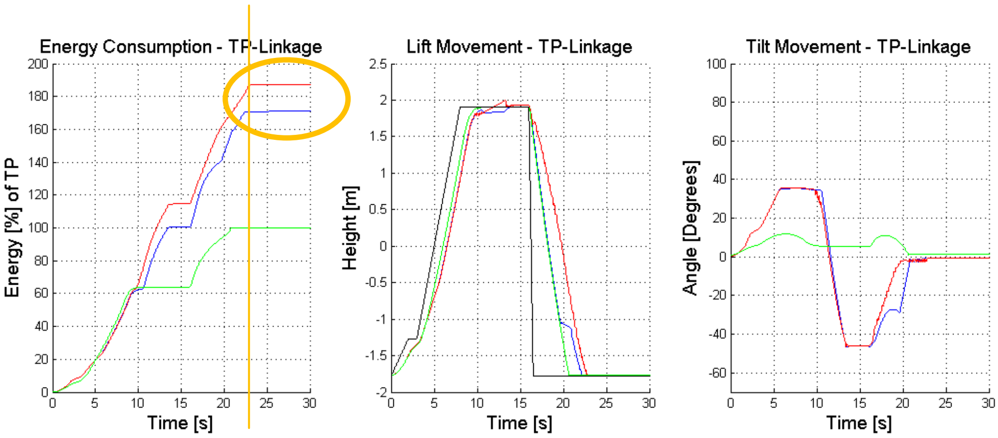

Figure 8.2 The energy consumption and movement for TP-linkage during the bucket work cycle. The green curves are the *reference case*, the blue is the *human case*, and the red is the *electro-hydraulic compensated case*.

The oscillations seen for the pressures in the TP and Z linkages give a reason to worry about if the models of these linkages are good enough.

The way that the linkage is controlled using both lift and tilt cylinder at the same time makes it possible for an oscillative behavior to rise if the regulators and system are not damped enough.

Oscillations could happen when the lift cylinder follows the reference curve and expands. This will change the tilt angle since they are coupled. If the regulator for the tilt angle cannot compensate the error without overshooting the following will happen. The tilt regulator compensates too much giving the tilt cylinder a huge force. This force could be big enough to actually bend the linkage down a bit, increasing the error for the lift height. This in turn would increase the force in the lift cylinder more, and change the tilt angle even more. This could lead to an oscillation in the system with the cylinders working against each other.

The high pressures on the plus side of the lift cylinders for TP and Z linkages in Figure 8.3 at t=10-15 seconds are unexpected and should be lower compared with measured data in Appendix C. It is probably a result from the oscillations explained earlier i.e. how the movement is controlled, since the minus side of the tilt cylinder also is high during that period of time. The two cylinders clearly work against each other resulting in that the pressure rises to maximum. The tilt cylinder is emptying the bucket at about t=10 seconds and will start the oscillation for the TP and Z linkages. Another source of the oscillations could be that the valves are too fast for the regulators.

The models have probably been built correct but would need to be better tuned and/or added friction to avoid the oscillations. The oscillations in the pressure does not necessarily increase the energy consumption and it might even be neglectable small. Since the linkage movement is not really affected by the oscillations it is safe to trust the results according to the author. No oscillations can be viewed in Figure 8.2.

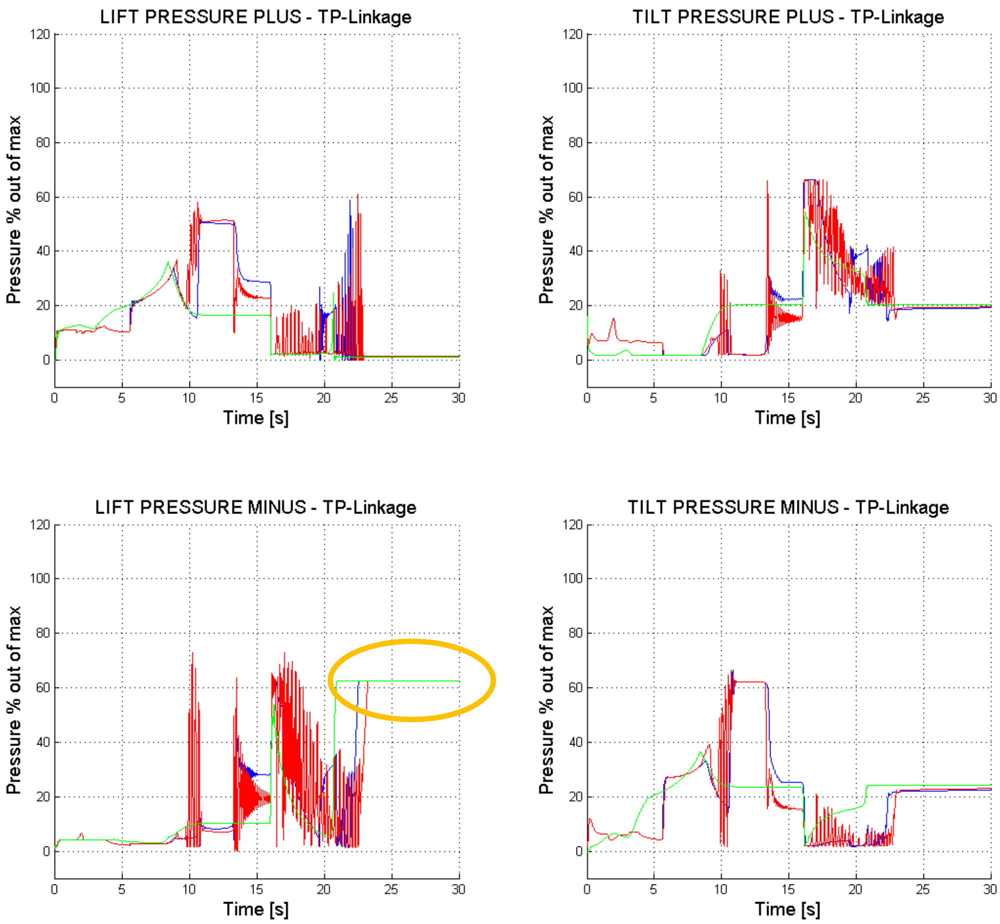

Figure 8.3 The cylinder pressures for TP-linkage during the bucket work cycle.

8.3.3 Fork work cycle without load

The TPC linkage has the lowest energy consumption when running the fork work cycle and it is also fastest when it comes to lowering the linkage.

It is worth to notice that the TPC almost needed the most energy to follow the bucket work cycle and least to follow the fork work cycle. The author would think this has to do with the geometry of the linkage and the fast displacements that the tilt cylinder has to do when emptying the bucket and lifting it up again. For the fork work cycle it also has the smallest and slowest displacement for the tilt cylinder. Large displacements demand a big amount of hydraulic fluid which increases the total energy consumption.

The fact that the pressure does not oscillate that much for the fork work cycle is probably a result of a much smoother movement of the linkage. The pressure curves look cleaner and are more reasonable compared to the measured data in Figure C.3 and Figure C.4. As for the bucket work cycle a high pressure sometimes gets locked inside the cylinders at the end of the cycle.

The oscillations do not seem to affect the energy consumption in this case, so the model is considered valid.

8.3.4 Bucket work cycle with 6 ton load

Since the linkage oscillates so much it probably means that the regulator model needs to be improved, either by better regulator parameters, slower valves or friction added to the mechanical system. However it is still likely that the energy consumption of this simulation shows the right tendencies. Since no abnormal behavior is shown in the energy consumption curve Figure F.1 compared to Figure D.1 when not having a load. They behave similarly.

8.3.5 Fork work cycle with 6 ton load

The results from the fork work cycle with a 6 ton load oscillated. Not all the linkages fulfilled the tilt angle requirements of a maximum of 5 degrees error. This means that the regulators do not work as good as they should. Comparing the results from the Z linkage for running the simulations with, Figure 8.4, and without the load, Figure 8.5, we can state that the behavior of the energy consumption curve is about the same for both of them. Since the tendencies are the same for the two simulations then probably the results have the right magnitude even if there are oscillations.

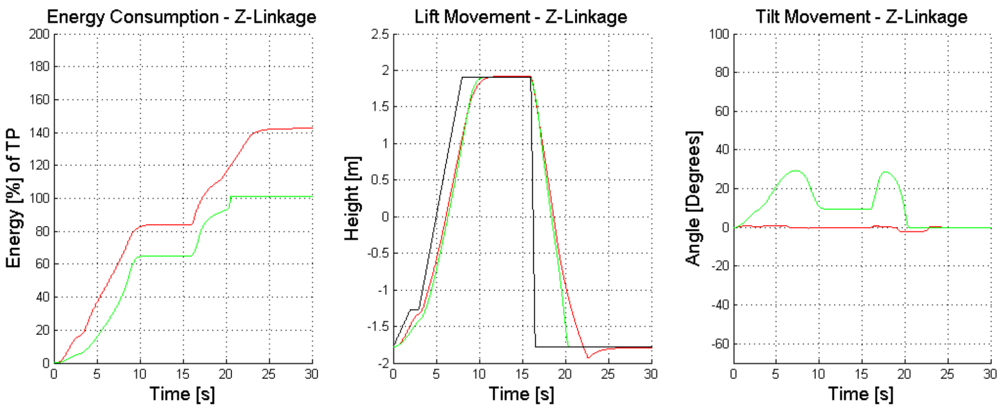

Figure 8.4 Energy consumption of Z linkage for fork work cycle without a load.

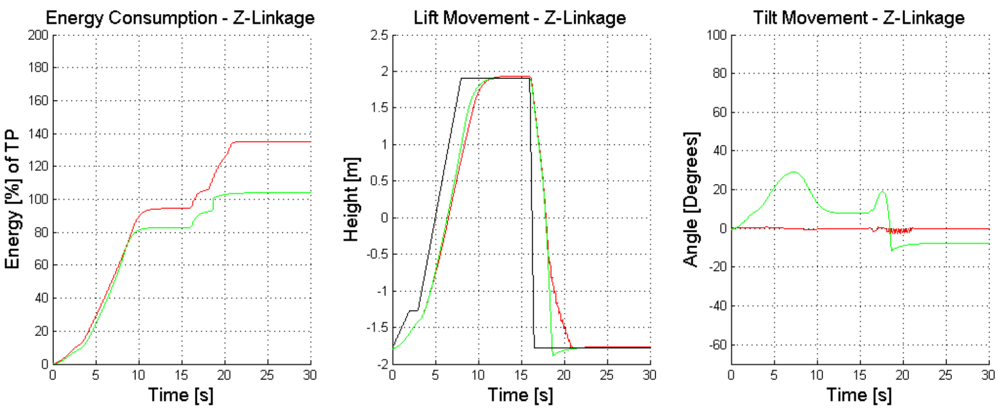

Figure 8.5 Energy consumption of Z linkage for fork work cycle with a load.

Since not all the linkages were able to fall freely it might be interesting to reduce the energy consumption for those linkages by the amount that was added from the pump when lowering the linkages. This could be a way of getting more correct results without changing the models.

8.4 Cost

The cost calculations from a TCO perspective show that ***Linkage 2*** was the best option for almost the whole spectrum. Not many WL are only used for lifting pallets. The results point out what linkage is most beneficial from a lifetime perspective of the linkage rather than what would be the cheapest to manufacture. The result should be used as a strong indicator of what linkage is best to use in a TCO perspective rather than providing exact numbers.

8.5 Lessons learned

It would have been better to start off by planning the control of the valves (that controls the linkage) more thoroughly in the beginning compared to what was done. The reason why it was not planned very well in the beginning was because of an underestimation of the need, of an advanced control. This led to that the control system was re-built a couple of times together with tuning of the control parameters. It could have saved some time to just plan ahead better.

The control of the pump should have been made infinite fast in the beginning and then a rate limit should have been added for it. This would have avoided other control problems that followed during the work.

It might have been better to have two different regulators for the bucket work cycle and the fork work cycle. The one for the bucket work cycle should represent how a human runs the machine and the fork should represent how a computer runs the machine. Although a refined human case was made for the bucket work cycle for this reason.

It might be better to only use one software so co-simulation problems such as; what time step to use and initial value problems would not occur. Probably it would be better to only use *SimuLink* since it is a more flexible software.

8.6 Problems

In the beginning a major problem with oscillations in the hydraulic system occurred. Partly since the design of the control was not good enough, and probably because friction is not included in the models. I think that friction in the model would have led to a more stable system.

The first design of an LS-system had some problems. An initial value made the pump have full displacement in the beginning of each simulation. That was solved with some clever redesign of the initial parameters.

It was not possible to simulate the system with a full load or to empty the bucket. So a true bucket work cycle was not possible to simulate.

8.7 Future of simulations

Modeling and simulations is a powerful tool and is an effective way of finding out if a new concept has any problems. It is often faster and cheaper than to build a prototype and will probably be used more in the future.

Chapter 9 – Conclusions

9.1 Conclusions

The TPC is the lightest linkage and TP is the heaviest. TP has the best parallel-adjustment and TPC needs to be continuously controlled since it does not have a self-adjustment of the tilt angle.

The design of the hydraulic system as well as how the movement of the linkage is controlled play a major part of how big the energy consumption gets together with the design of the linkage. It is better to have the angle dependent of lift height than dependent of time, while simulating.

When comparing the results to measured values it shows that the system behaves more or less as expected, only a bit too much oscillations. The results are reasonable and give a good indication of what linkage is best suitable even though some results had a lot of oscillations. It also shows that the energy consumption using an electro-hydraulic compensation is small, compared to the unregulated work case, which only lifts and lowers the linkage.

Linkage 2 had the lowest cost from a TCO point regardless of how many hours the vehicle was used or at what fraction it was driven with forks contra bucket up to 90 %. ***Linkage 1*** had the highest cost throughout the whole spectrum.

9.2 Recommendations

The authors recomendation, from a TCO perspective, is to use ***Linkage 2*** on a WL since it showed to be the cheapest linkage as in Figure 9.1. Of course some consideration has to be taken to if the flat figures used to calculate the cost of the linkage are reasonable or not.

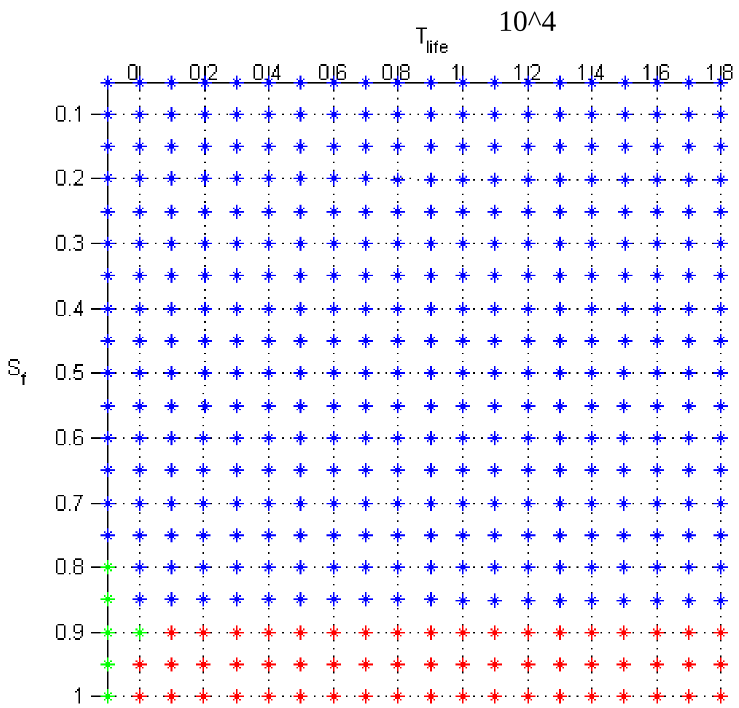

Figure 9.1 TCO results.

9.3 Future work

For future work there are a few things that would be good to proceed with. First of all, include the cylinder masses and their motion in the model, increasing the accuracy. Then add friction to the model to reduce the oscillations.

- The angle reference could be optimized for each linkage instead of running the same for all linkages. It might have been a good idea to make it move as the conclusions of [5] states, not to raise the center of mass higher than necessary.
- If oscillations remain after friction is added to the model, it is recommended to try to tune the regulator parameters better.
- It would be interesting to use the analysis presented in [4] to evaluate the linkages used in this report and compare to the results presented here.
- Make a more reliable cost analysis of the production of the linkage, time, material and production method.
- Investigate if there is any increase of productivity when using an electro-hydraulic parallel compensated linkage compared to not having one.
- Do measurements to validate the model.
- More advanced models that can handle emptying of a bucket.

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Appendix A Nomenclature of Hydraulic Components

These most common symbols used in hydraulics and in this report can be viewed in Figure A.1.

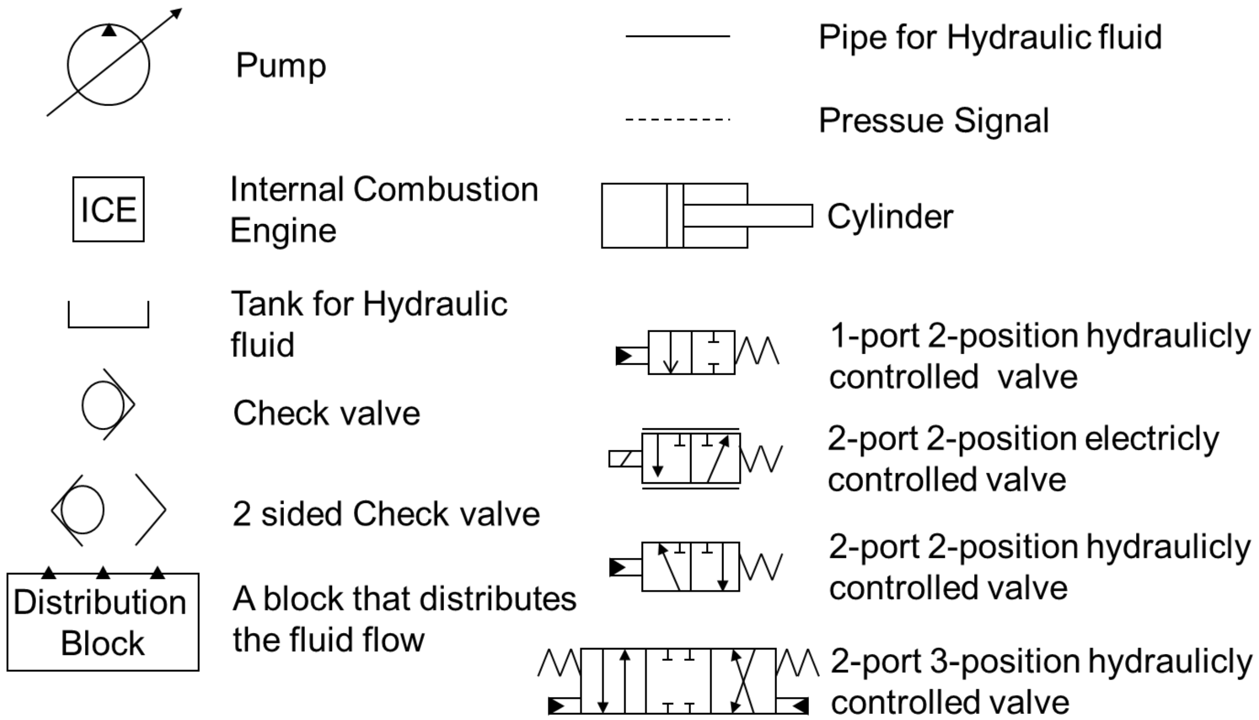

Figure A.1 Some common symbols of hydraulic components.

Appendix B Forces and Torques in a Linkage

By separating the links from the linkage in Figure B.1 and adding all the forces and torque that affects the linkage will result as in Figure B.2.

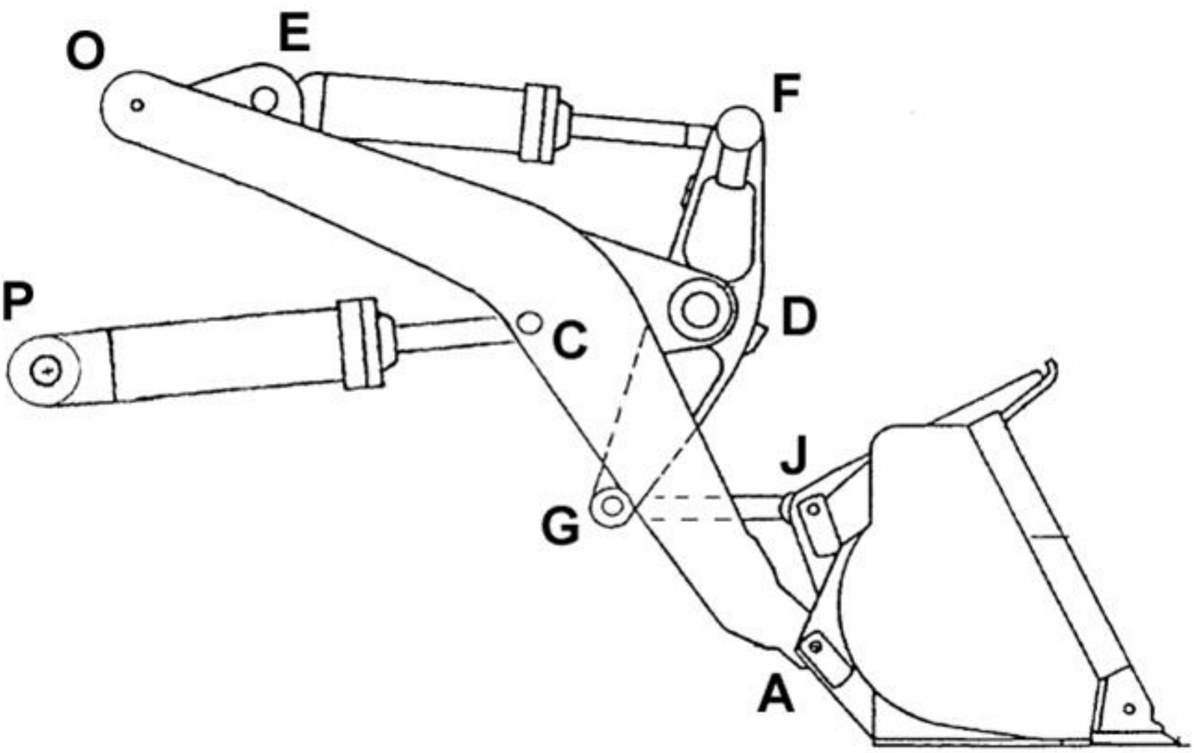

Figure B.1 Z-linkage.

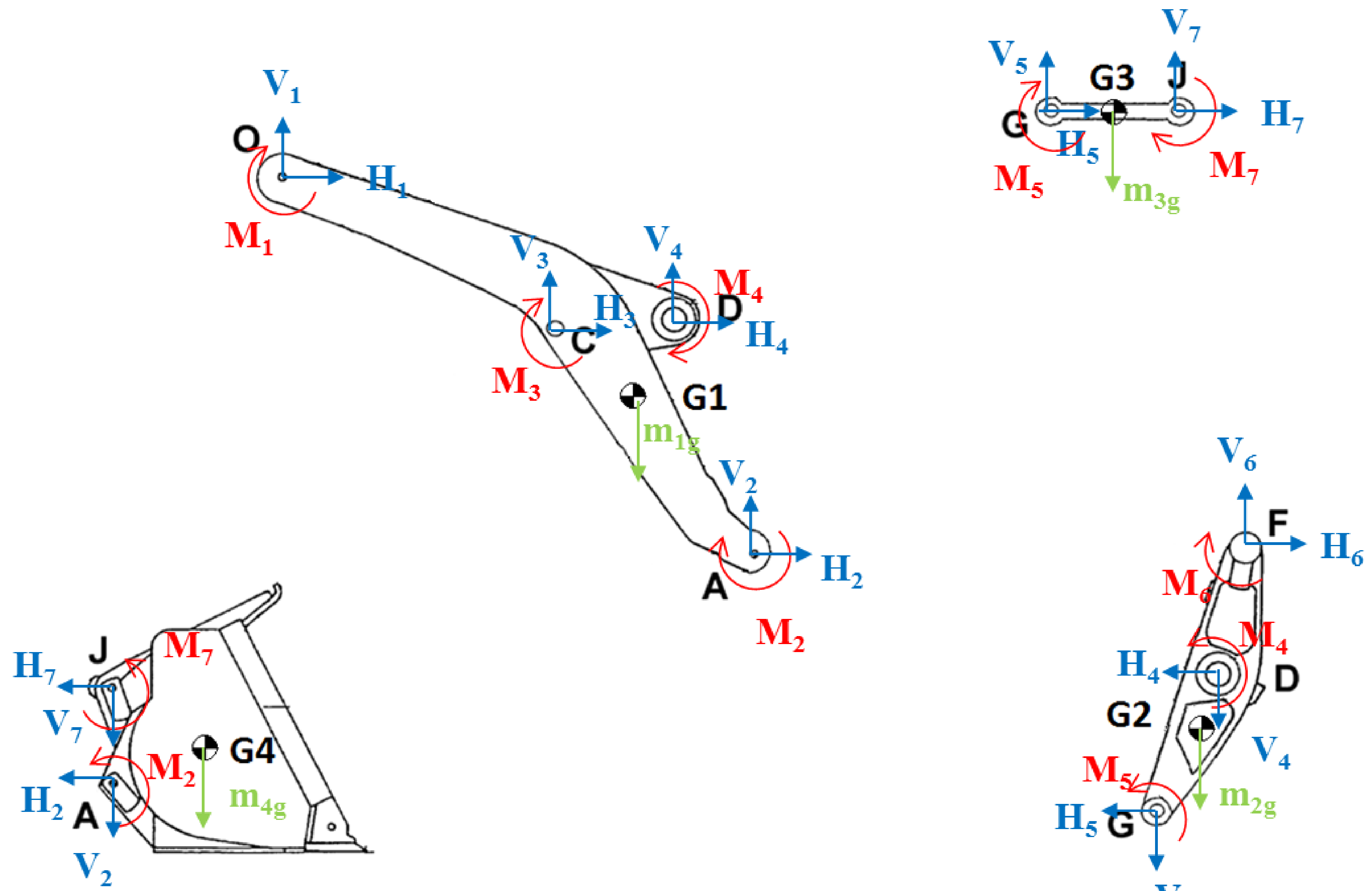

Figure B.2 Forces and torques working on the linkage.

Appendix C Measured Data

Measured data from three work cycles after each other has been provided by Volvo CE and is presented in this appendix. The displacement of the tilt cylinder can be viewed in Figure C.1 and the lift cylinder in Figure C.2. The pressures in the tilt cylinder can be viewed in Figure C.3 and the lift cylinder in Figure C.4.

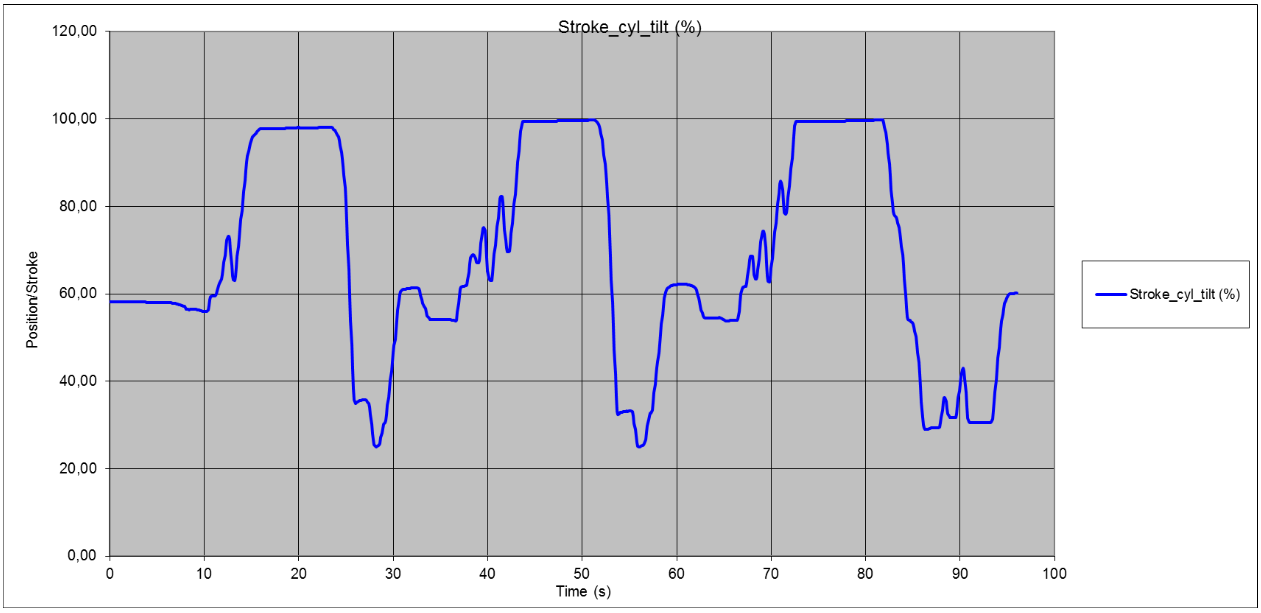

Figure C.1 Displacement of a tilt cylinder for three consecutive bucket work cycles.

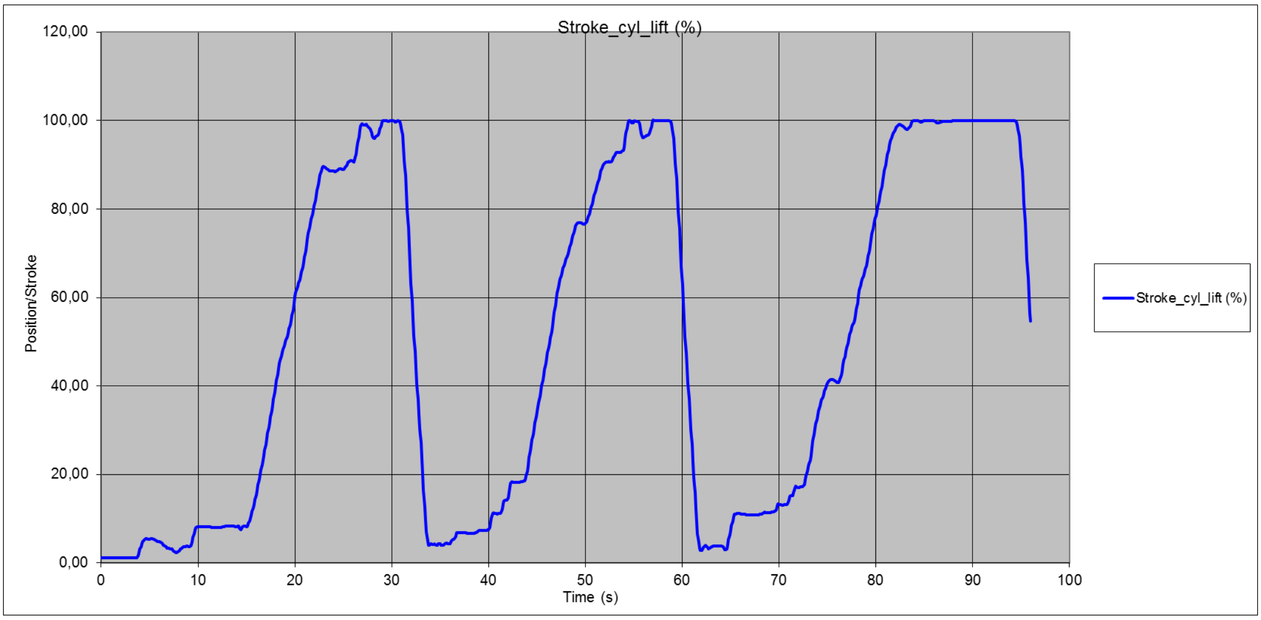

Figure C.2 Displacement of a lift cylinder for three consecutive bucket work cycles.

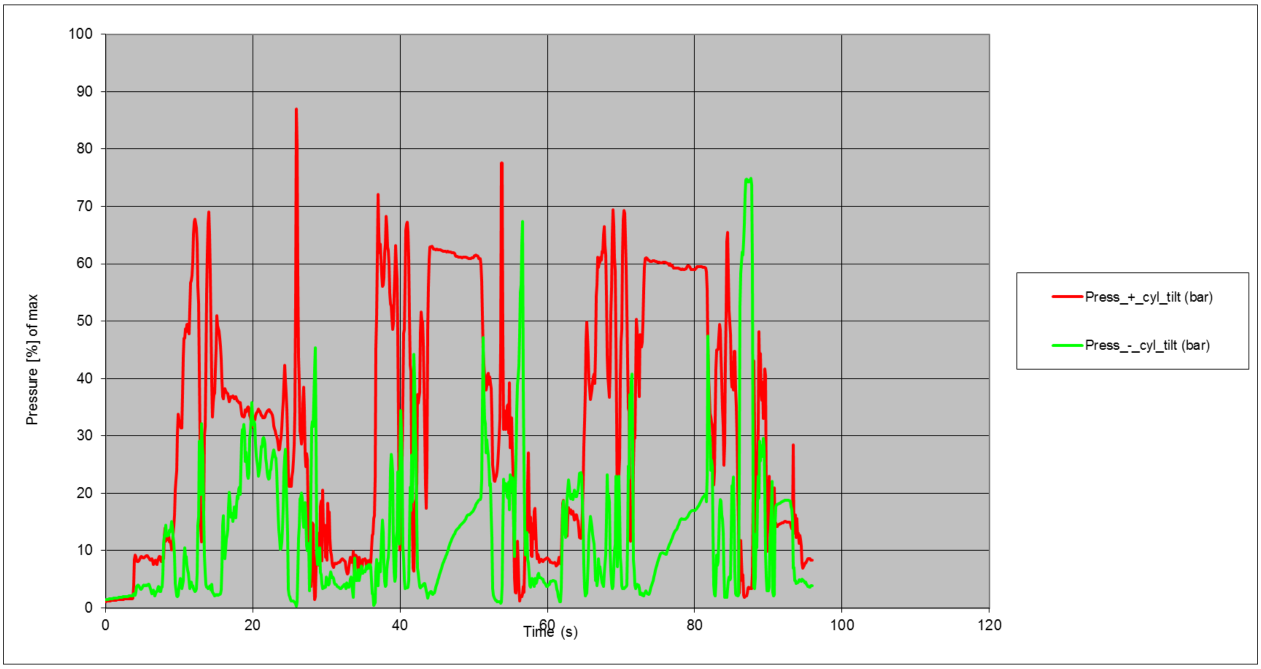

Figure C.3 Pressures of a tilt cylinder for three consecutive bucket work cycles.

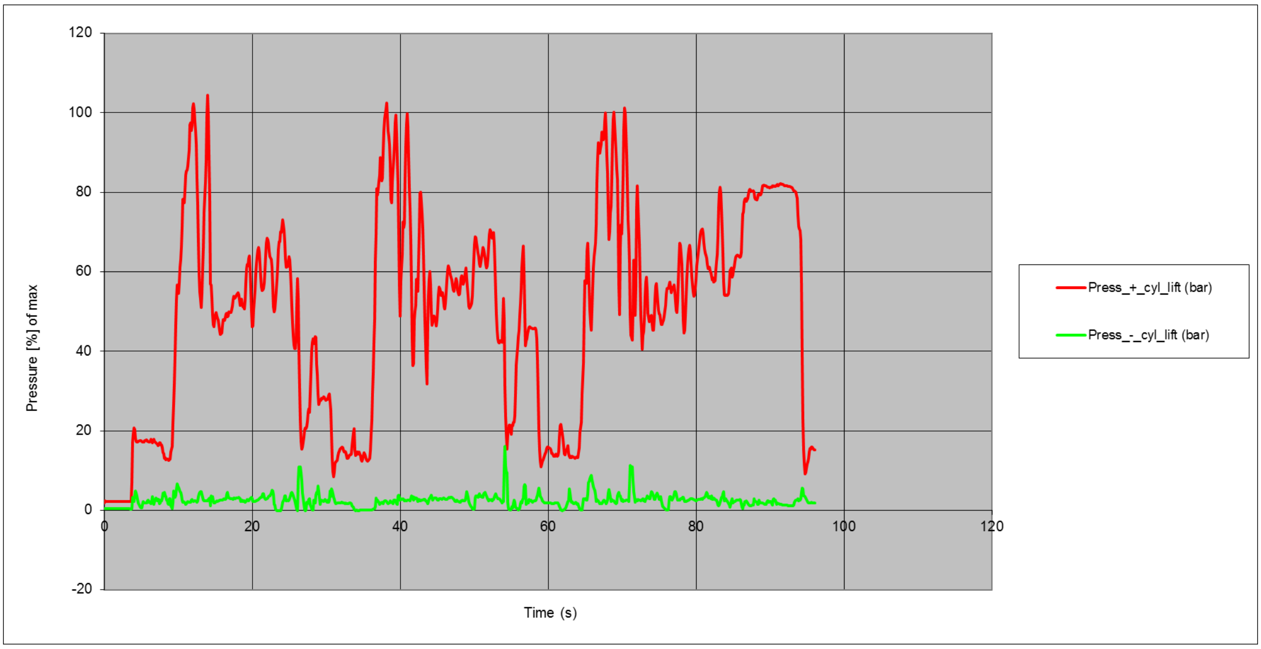

Figure C.4 Pressures of a lift cylinder for three consecutive bucket work cycles.

Appendix D Result of Bucket without Load

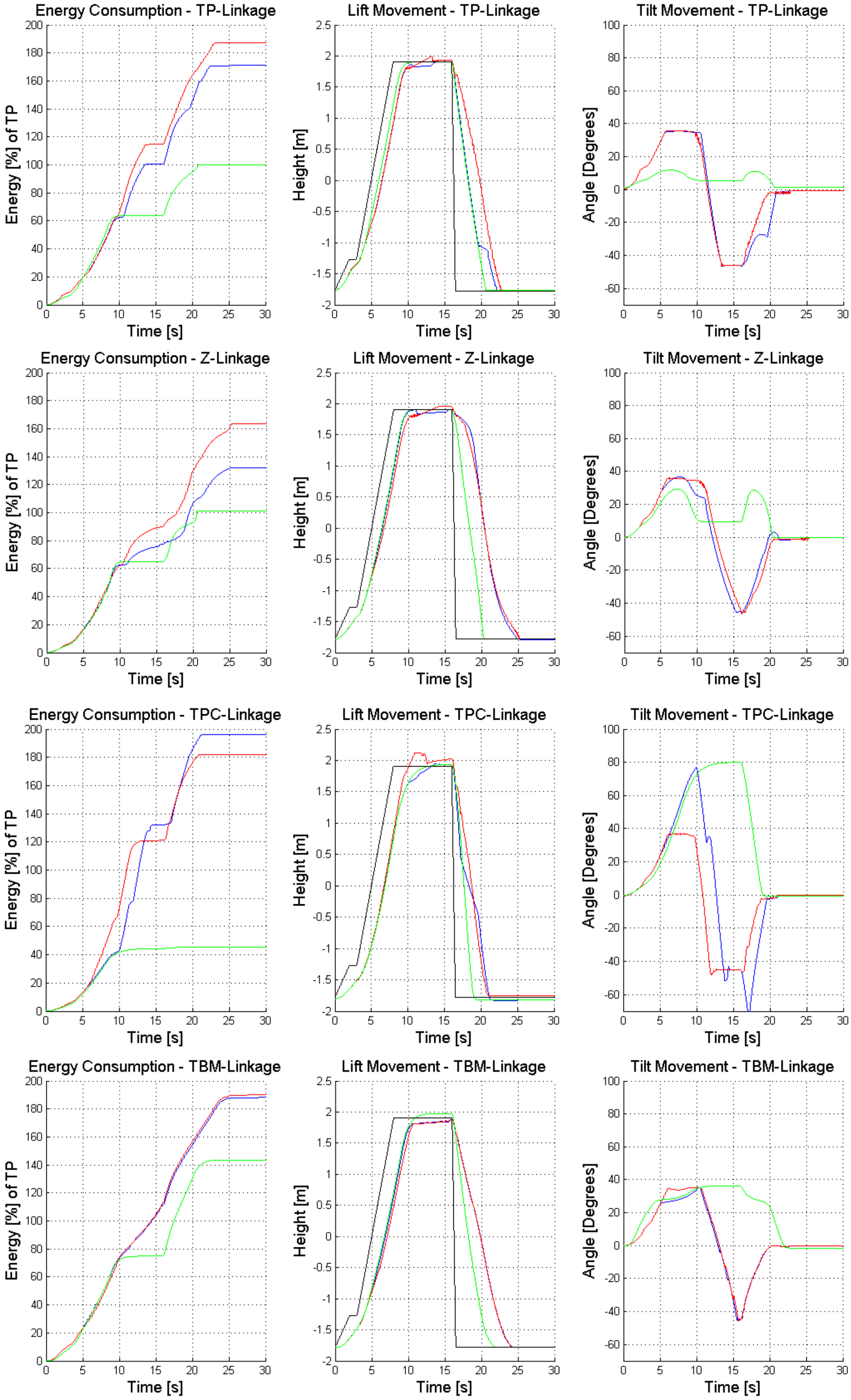

Figure D.1 Results from the bucket work cycle. Each row of the diagrams presents the result for a specific linkage. The diagram to the left shows the energy consumption normed with the TP linkage, the middle shows the height of the linkage (from O to A bearing) and the right diagram shows the tilt angle of the linkage, during the bucket work cycle.

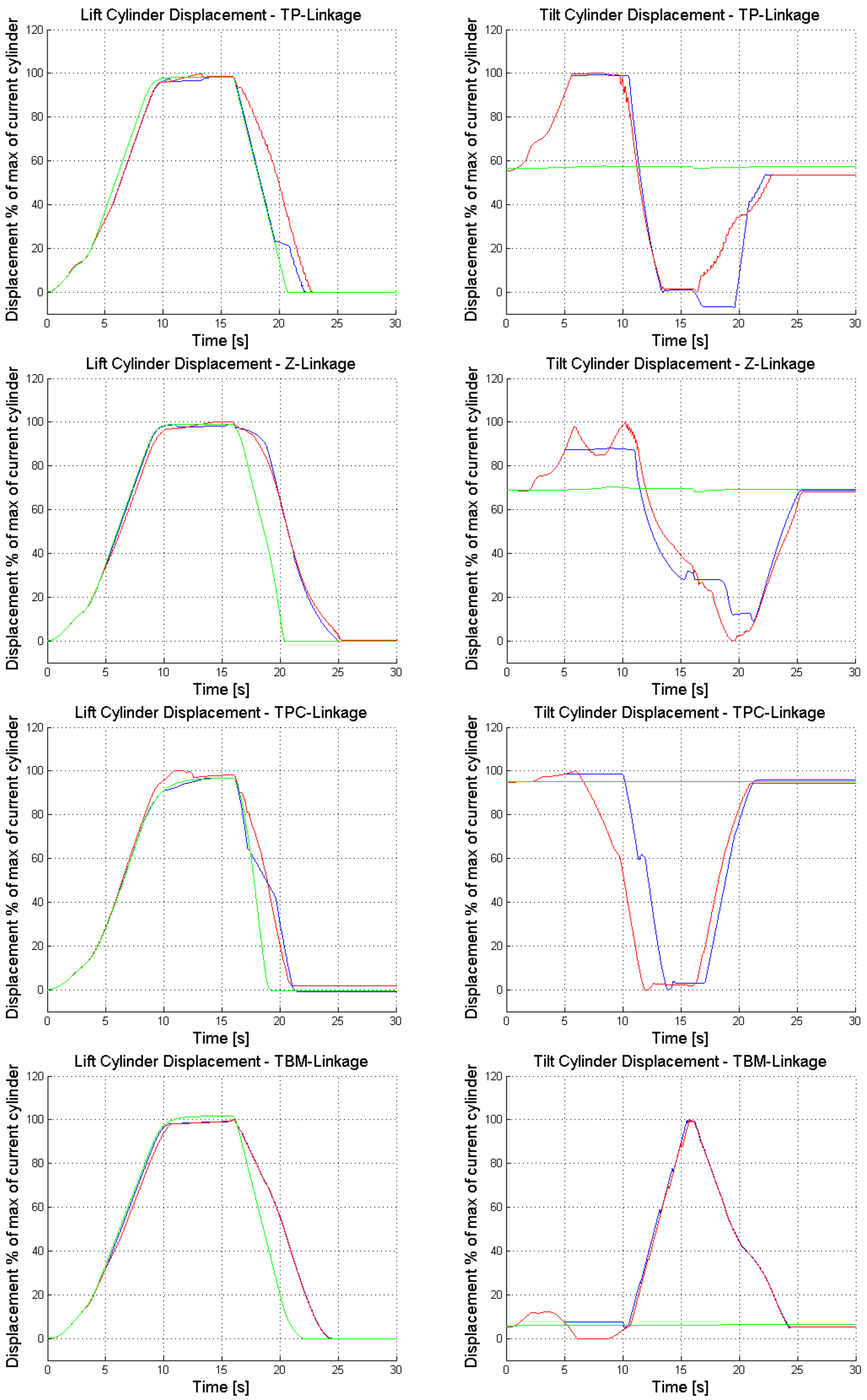

Figure D.2 The curves display the movement of the lift cylinder and the tilt cylinder of the different linkages. The curves are normalized with their own maximum movement, during the bucket work cycle.

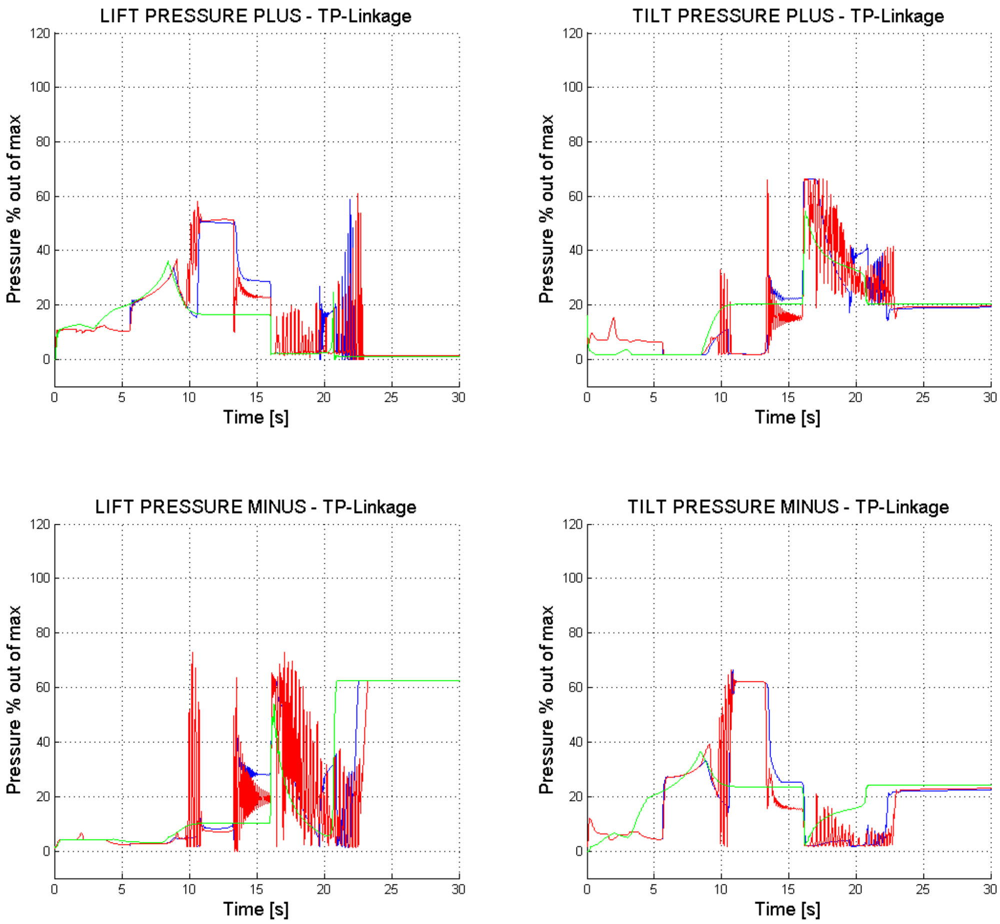

Figure D.3 The pressures on the + and - side of the lift and tilt cylinders for the TP-linkage during the bucket work cycle.

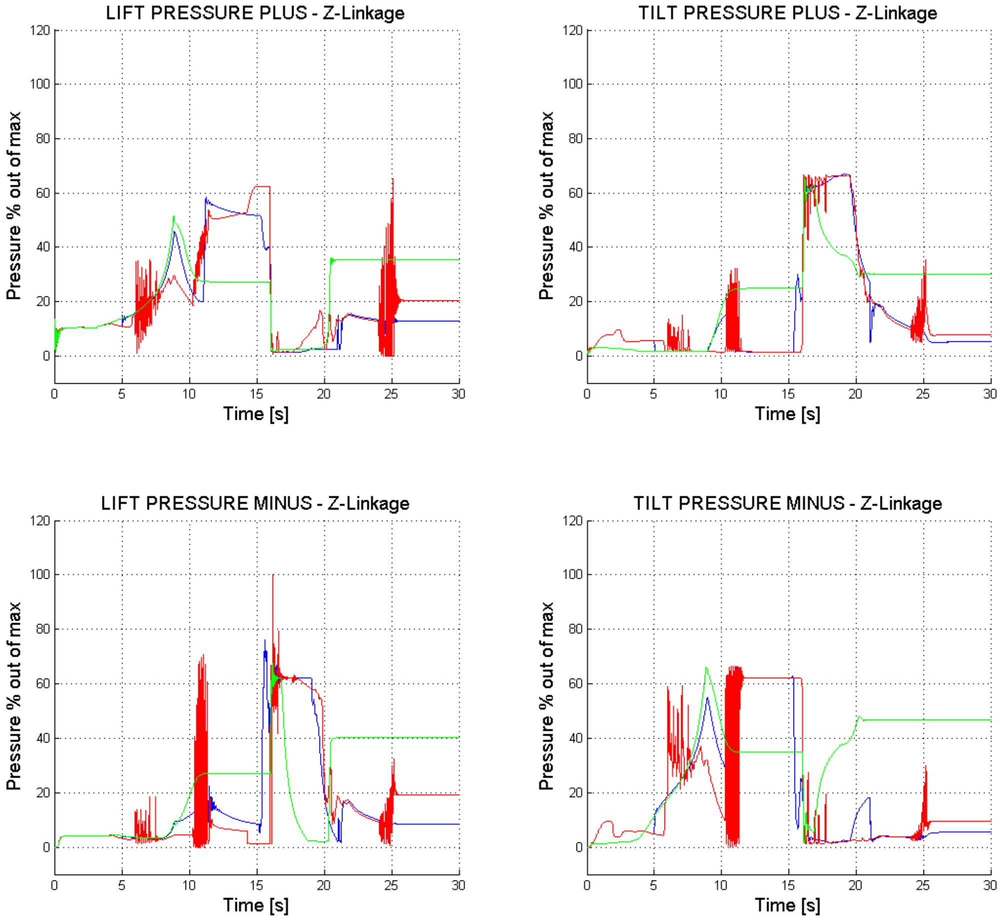

Figure D.4 The pressures on the + and - side of the lift and tilt cylinders for the Z-linkage during the bucket work cycle.

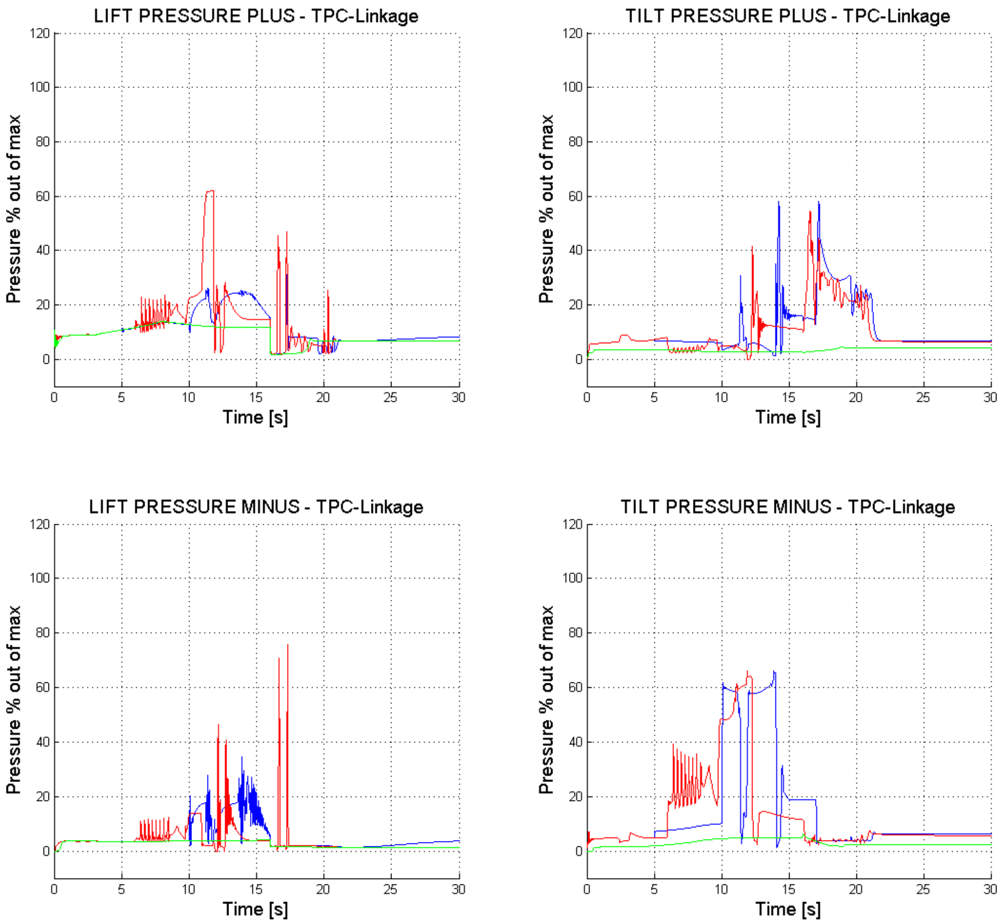

Figure D.5 The pressures on the + and - side of the lift and tilt cylinders for the TPC-linkage during the bucket work cycle.

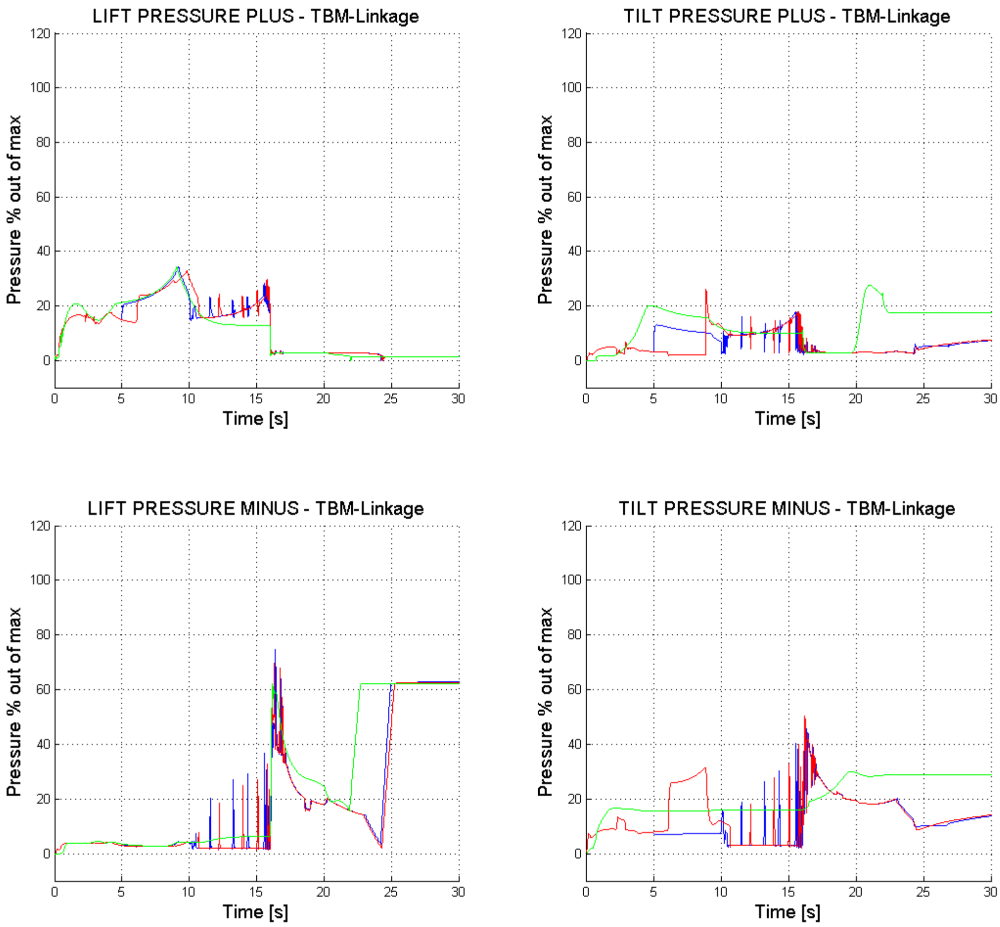

Figure D.6 The pressures on the + and - side of the lift and tilt cylinders for the TBM-linkage during the bucket work cycle.

Appendix E Result of Forks without Load

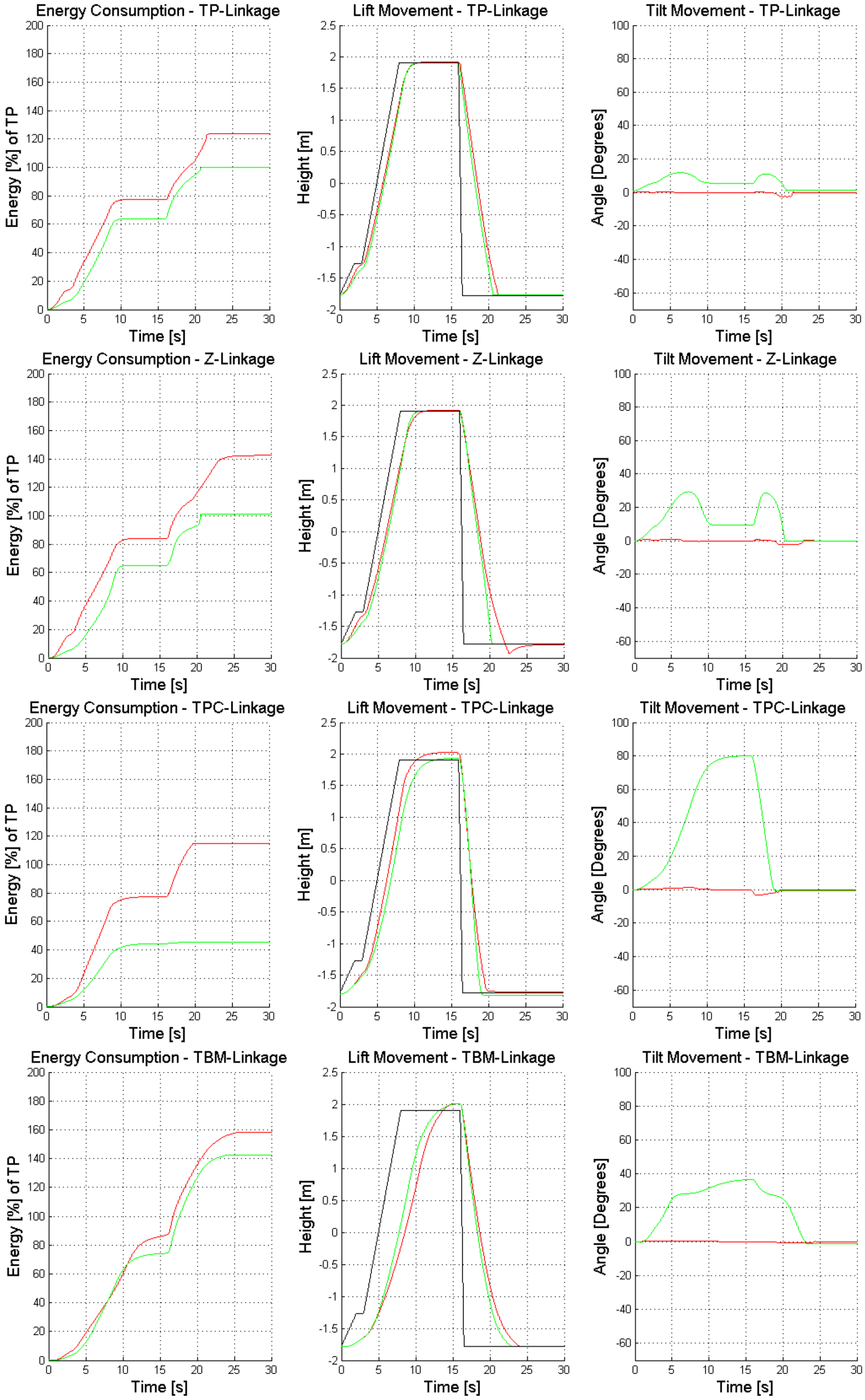

Figure E.1 Results from the fork work cycle. Each row of the diagrams presents the result for a specific linkage. The diagram to the left shows the energy consumption normed with the TP linkage, the middle shows the height of the linkage (from O to A bearing) and the right diagram shows the tilt angle of the linkage, during the fork work cycle.

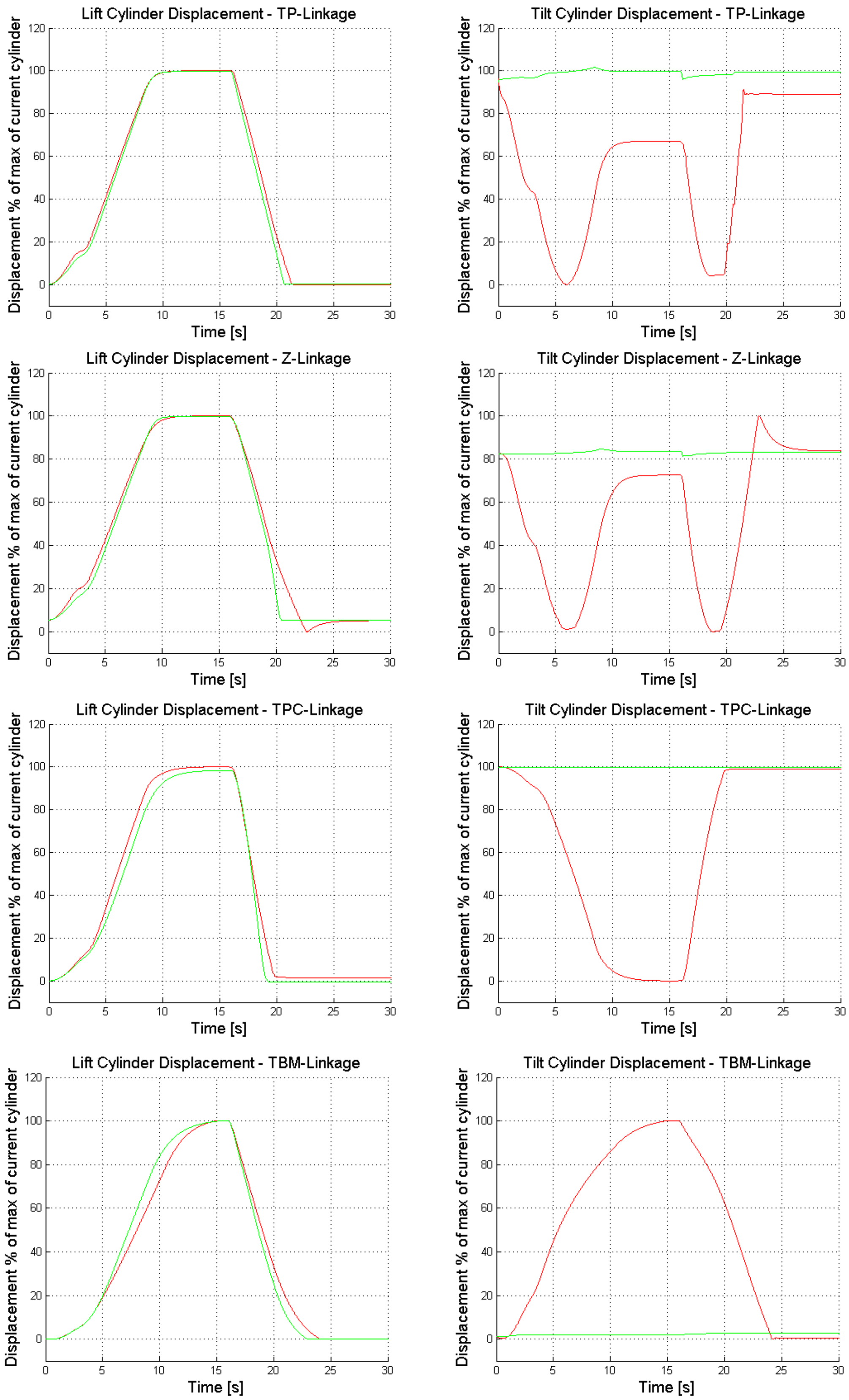

Figure E.2 The curves display the movement of the lift cylinder and the tilt cylinder of the different linkages. The curves are normalized with their own maximum movement, during the fork work cycle.

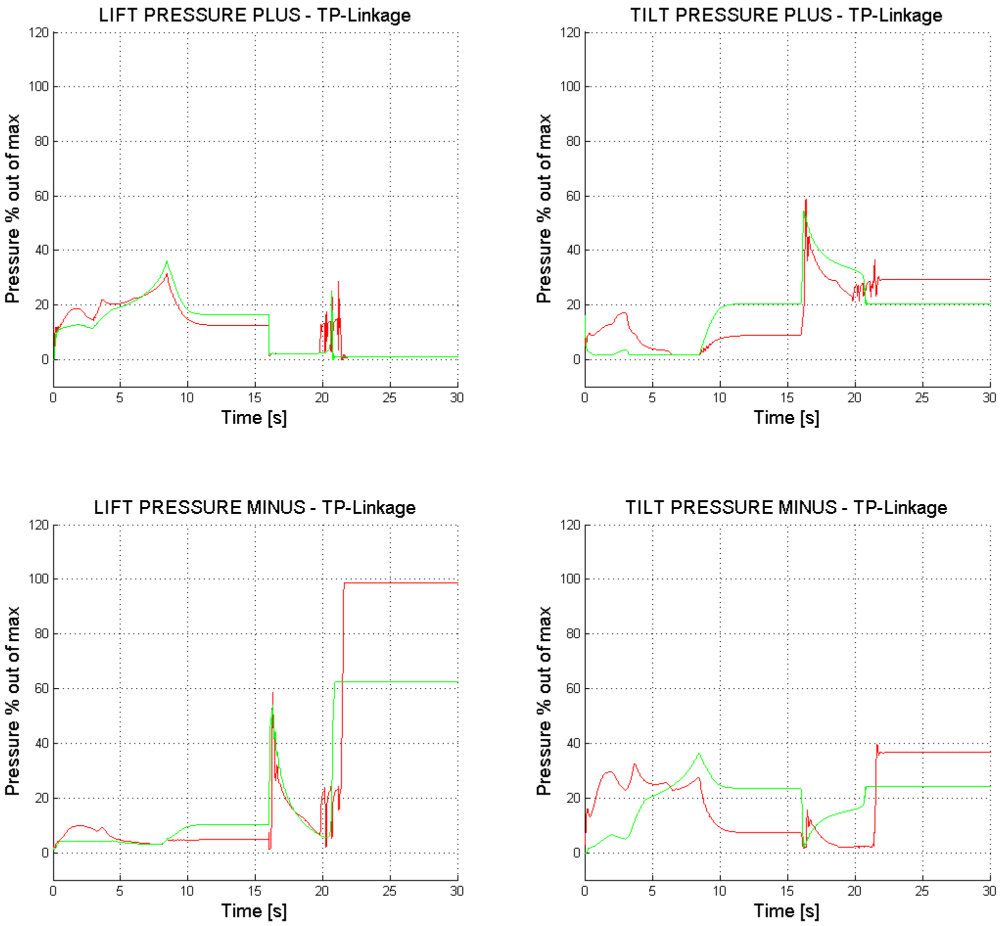

Figure E.3 The pressures on the + and - side of the lift and tilt cylinders for the TP-linkage during the during the fork work cycle.

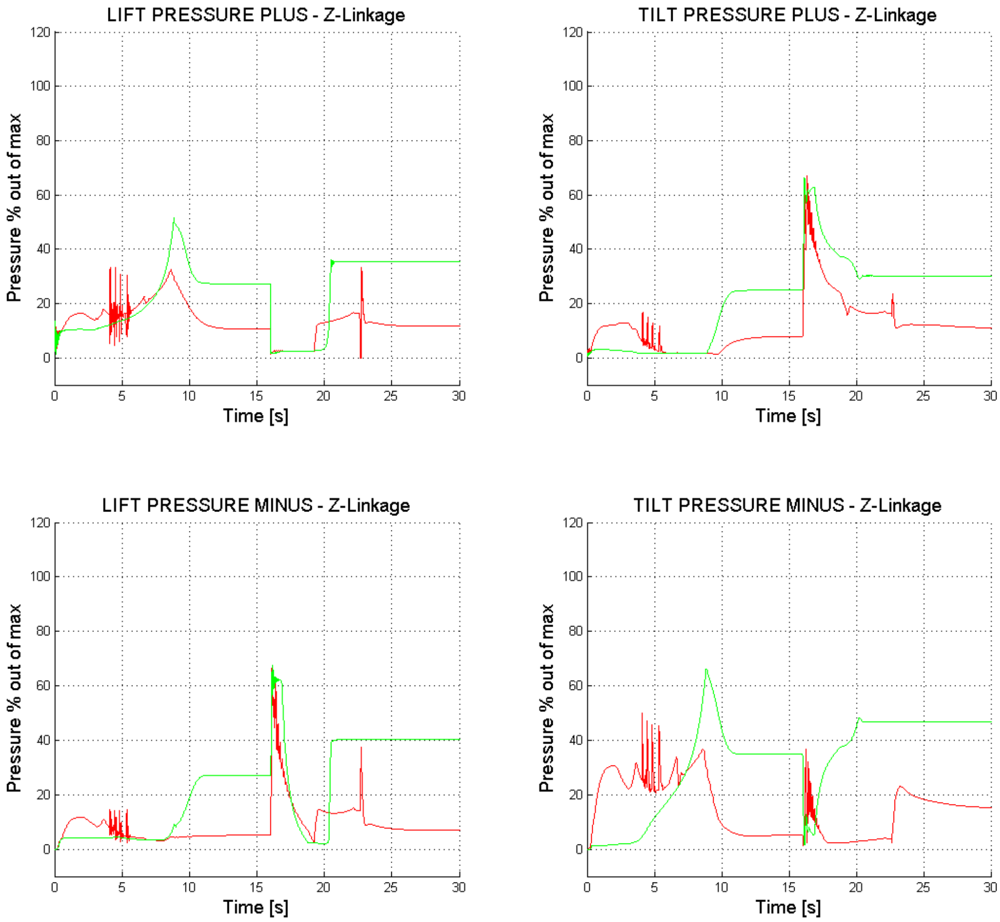

Figure E.4 The pressures on the + and - side of the lift and tilt cylinders for the Z-linkage during the fork work cycle.

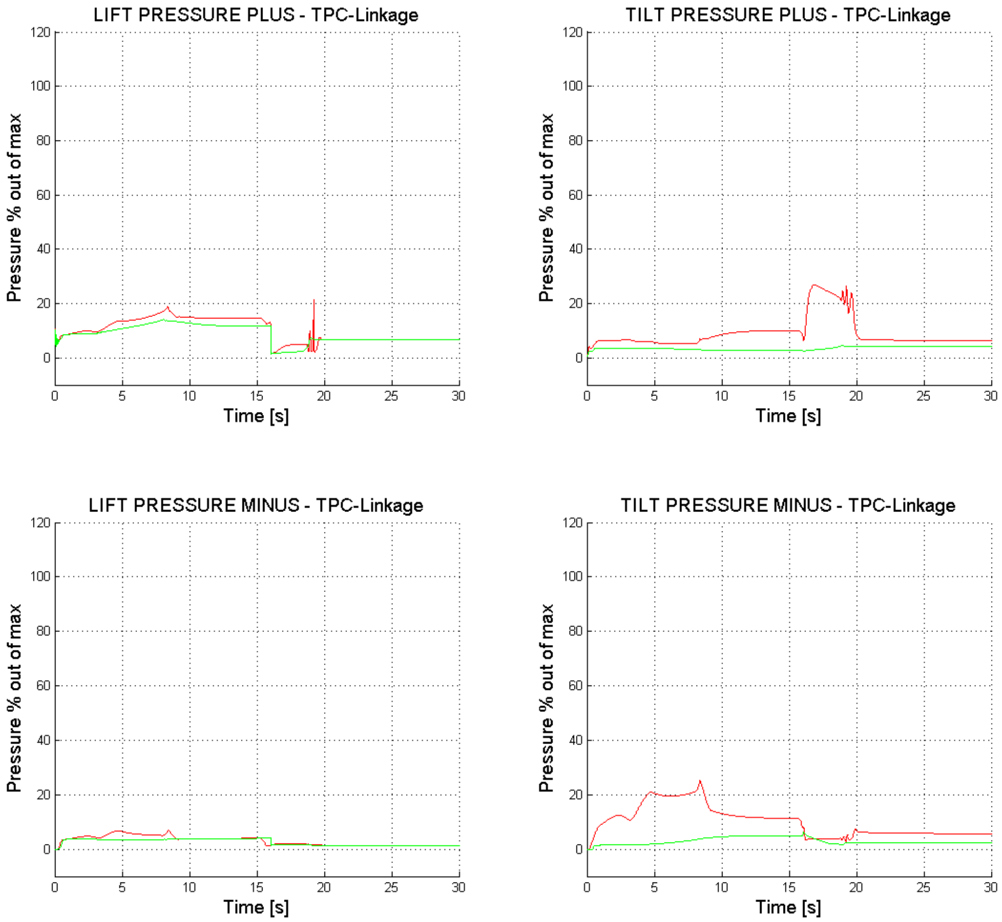

Figure E.5 The pressures on the + and - side of the lift and tilt cylinders for the TPC-linkage during the fork work cycle.

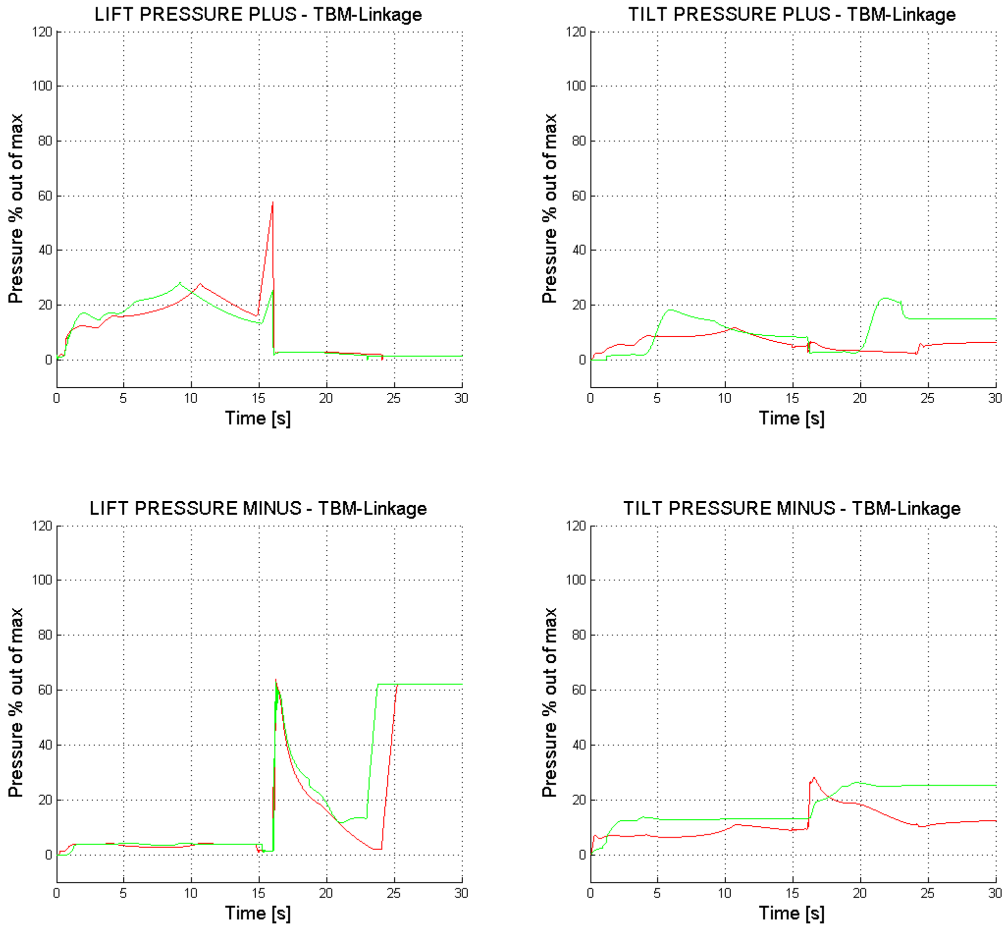

Figure E.6 The pressures on the + and - side of the lift and tilt cylinders for the TBM-linkage during the fork work cycle.

Appendix F Result of Bucket with a 6 ton Load

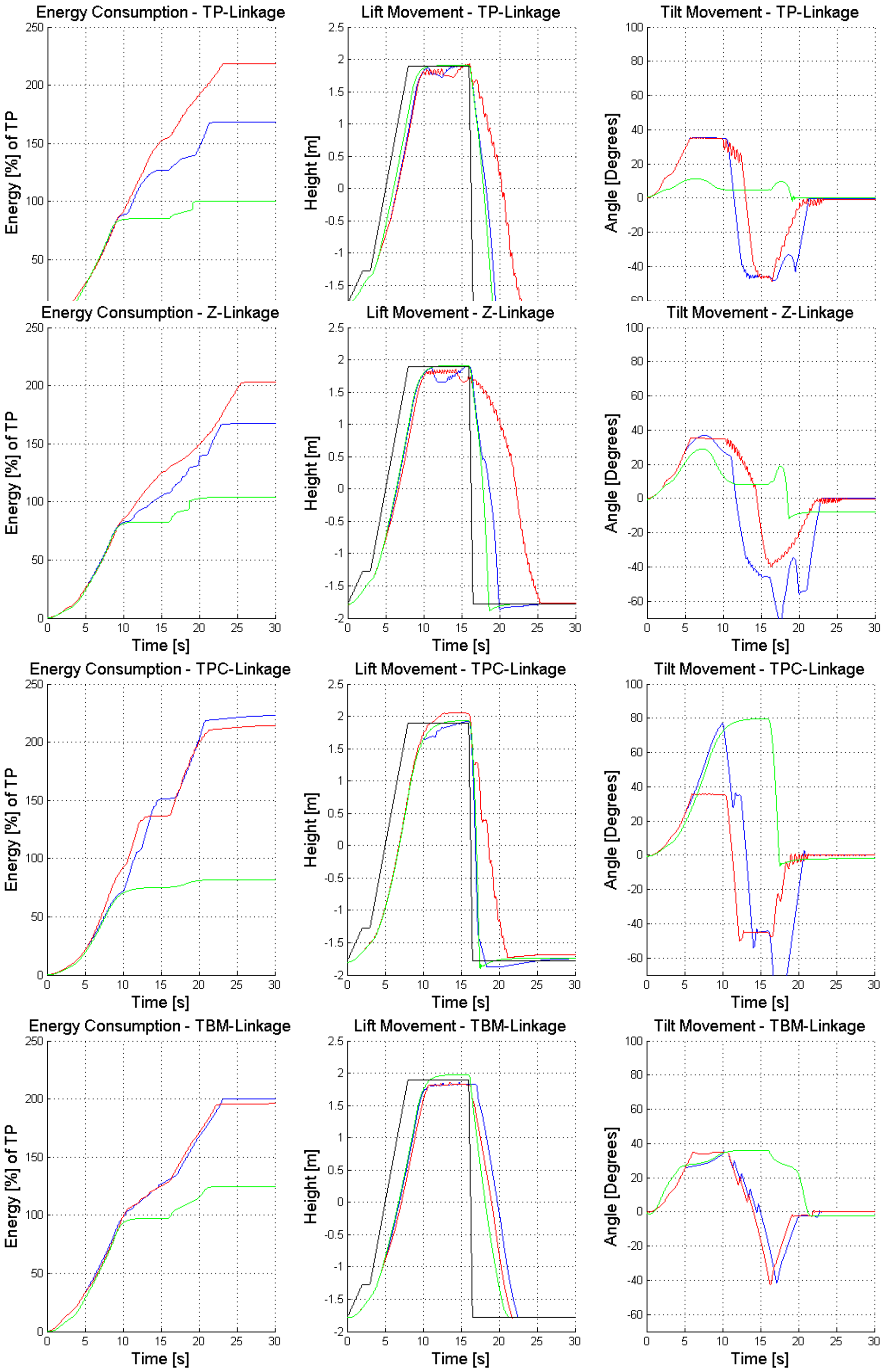

Figure F.1 Results from the bucket work cycle with a 6 ton load. Each row of the diagrams presents the result for a specific linkage. The diagram to the left shows the energy consumption normed with the TP linkage, the middle shows the height of the linkage (from O to A bearing) and the right diagram shows the tilt angle of the linkage, during the bucket work cycle.

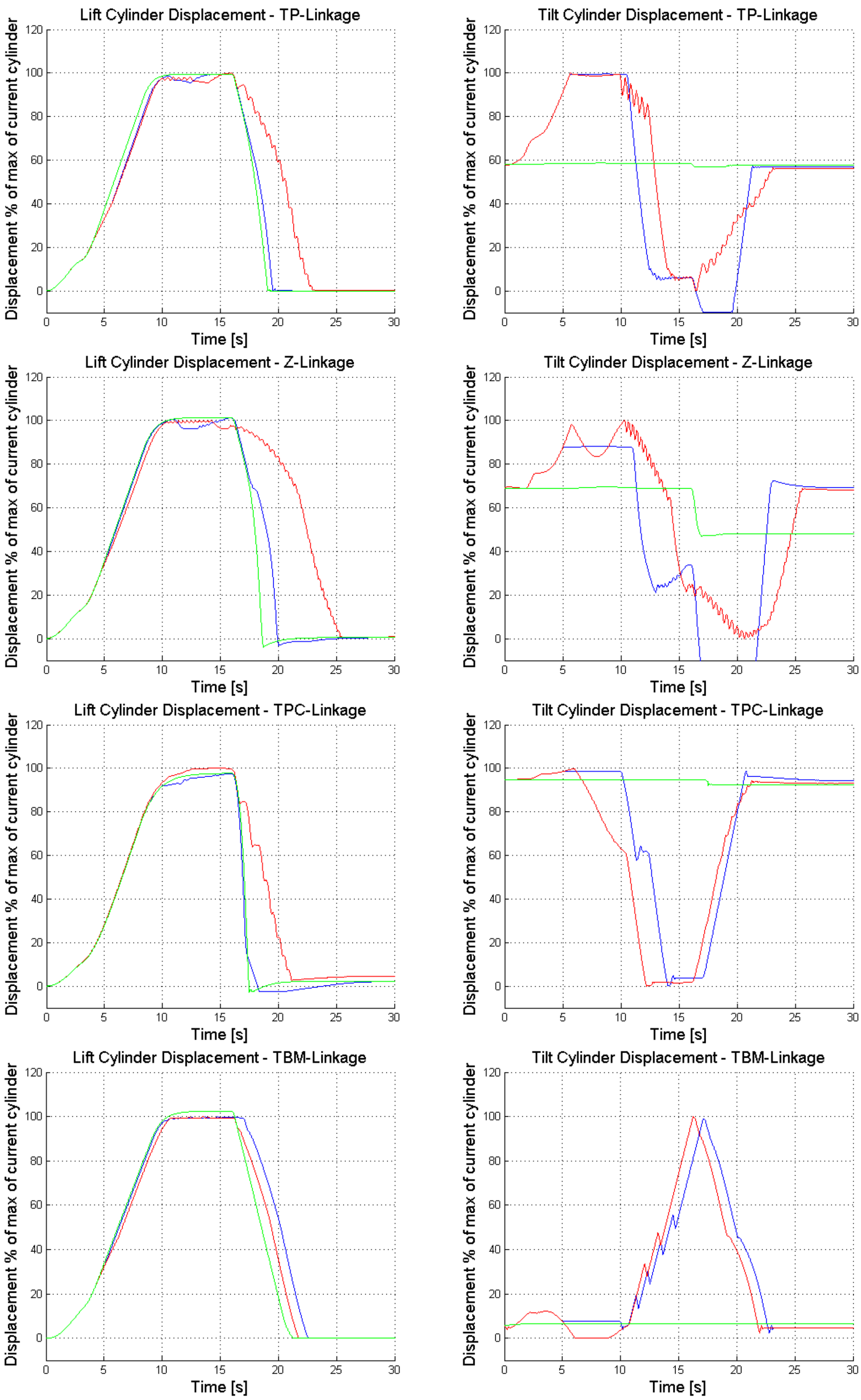

Figure F.2 The curves display the movement of the lift cylinder and the tilt cylinder of the different linkages. The curves are normalized with their own maximum movement, during the bucket work cycle with a 6 ton load.

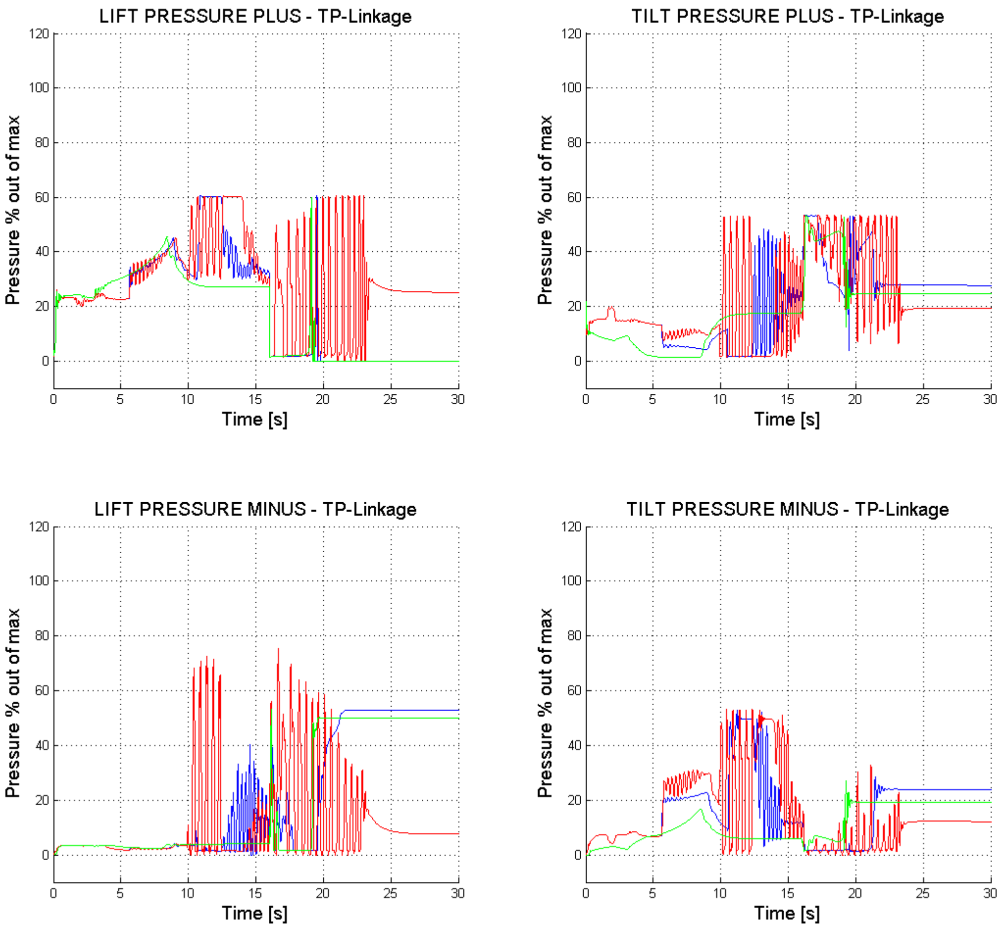

Figure F.3 The pressures on the + and - side of the lift and tilt cylinders for the TP-linkage during the bucket work cycle with a 6 ton load.

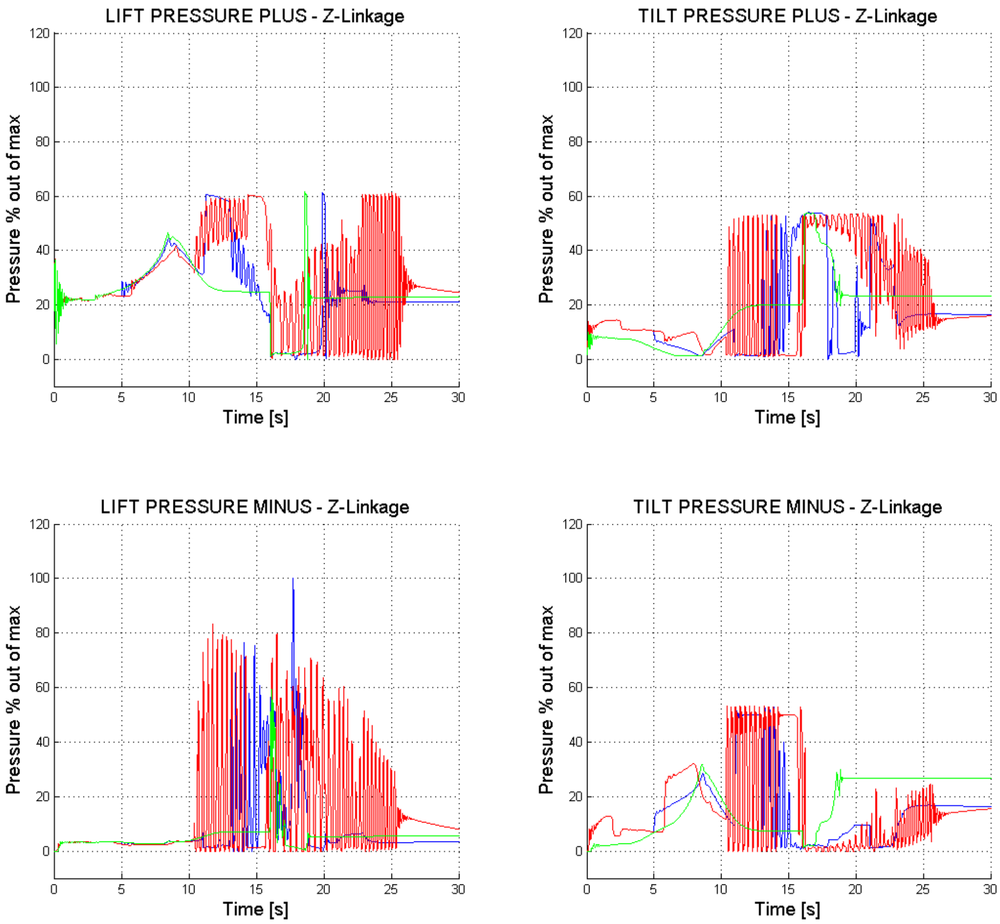

Figure F.4 The pressures on the + and - side of the lift and tilt cylinders for the Z-linkage during the bucket work cycle with a 6 ton load.

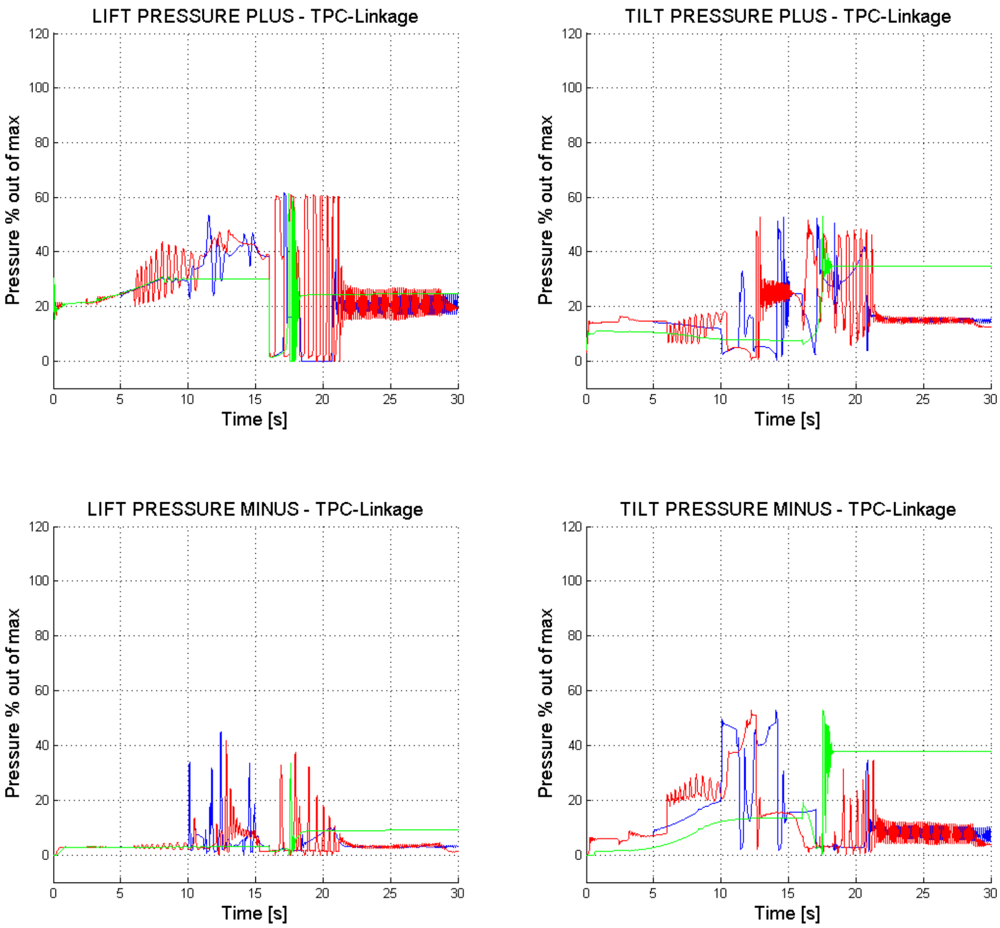

Figure F.5 The pressures on the + and - side of the lift and tilt cylinders for the TPC-linkage during the bucket work cycle with a 6 ton load.

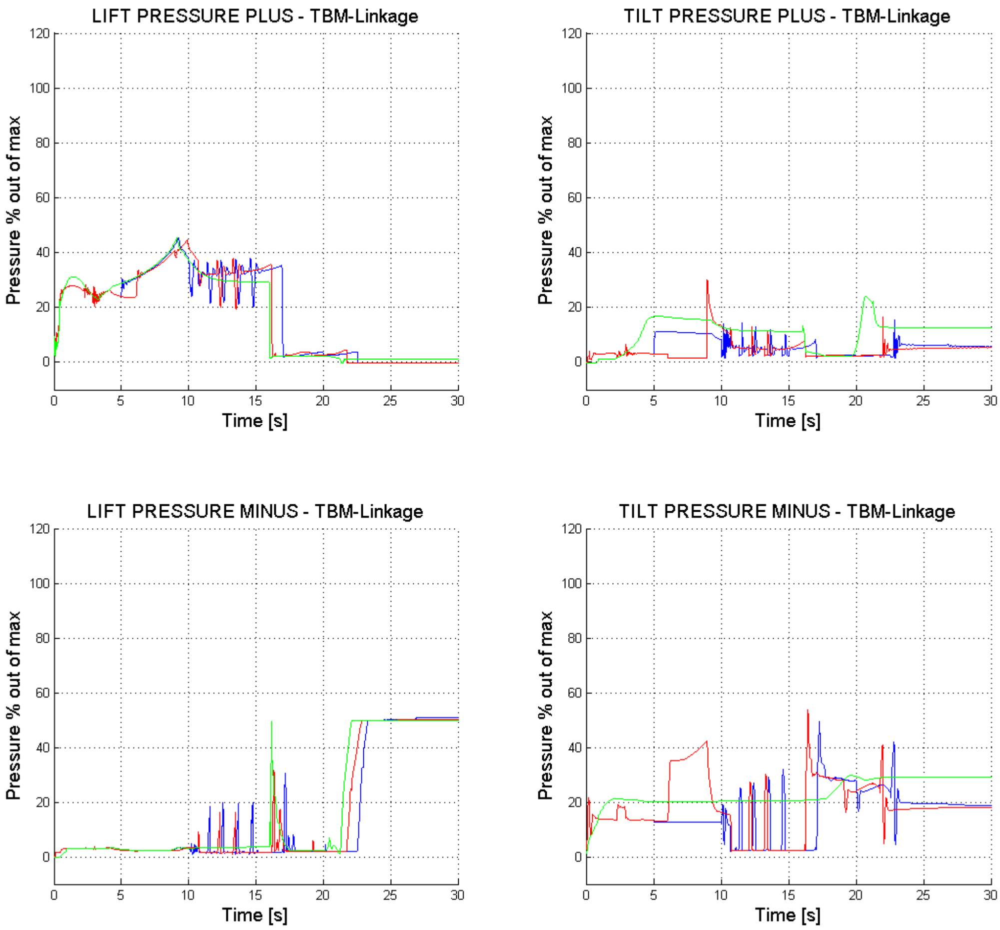

Figure F.6 The pressures on the + and - side of the lift and tilt cylinders for the TBM-linkage during the bucket work cycle with a 6 ton load.

Appendix G Result of Forks with a 6 ton Load

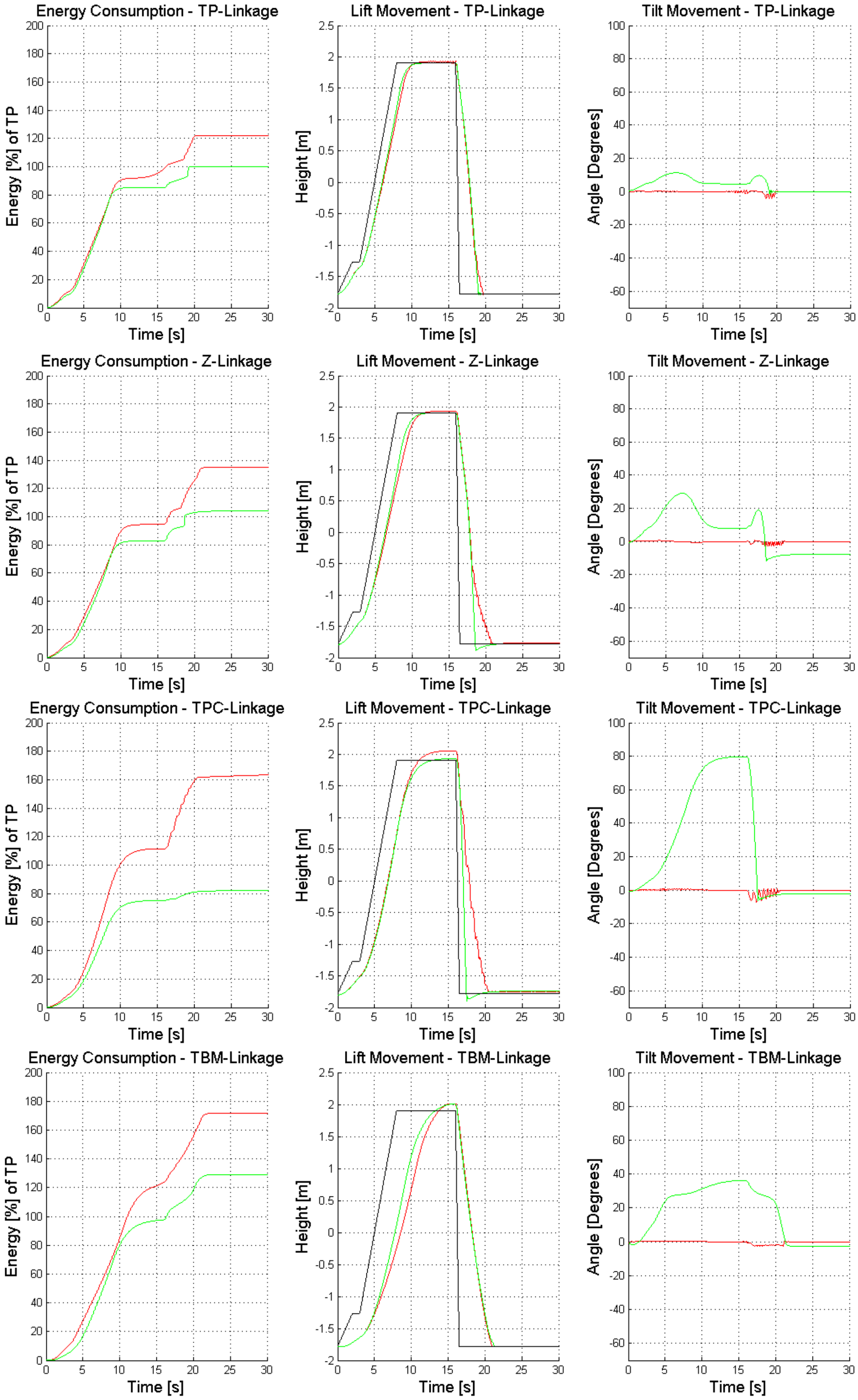

Figure G.1 Results from the fork work cycle. Each row of the diagrams presents the result for a specific linkage. The diagram to the left shows the energy consumption normed with the TP linkage, the middle shows the height of the linkage (from O to A bearing) and the right diagram shows the tilt angle of the linkage, during the fork work cycle with a 6 ton load.

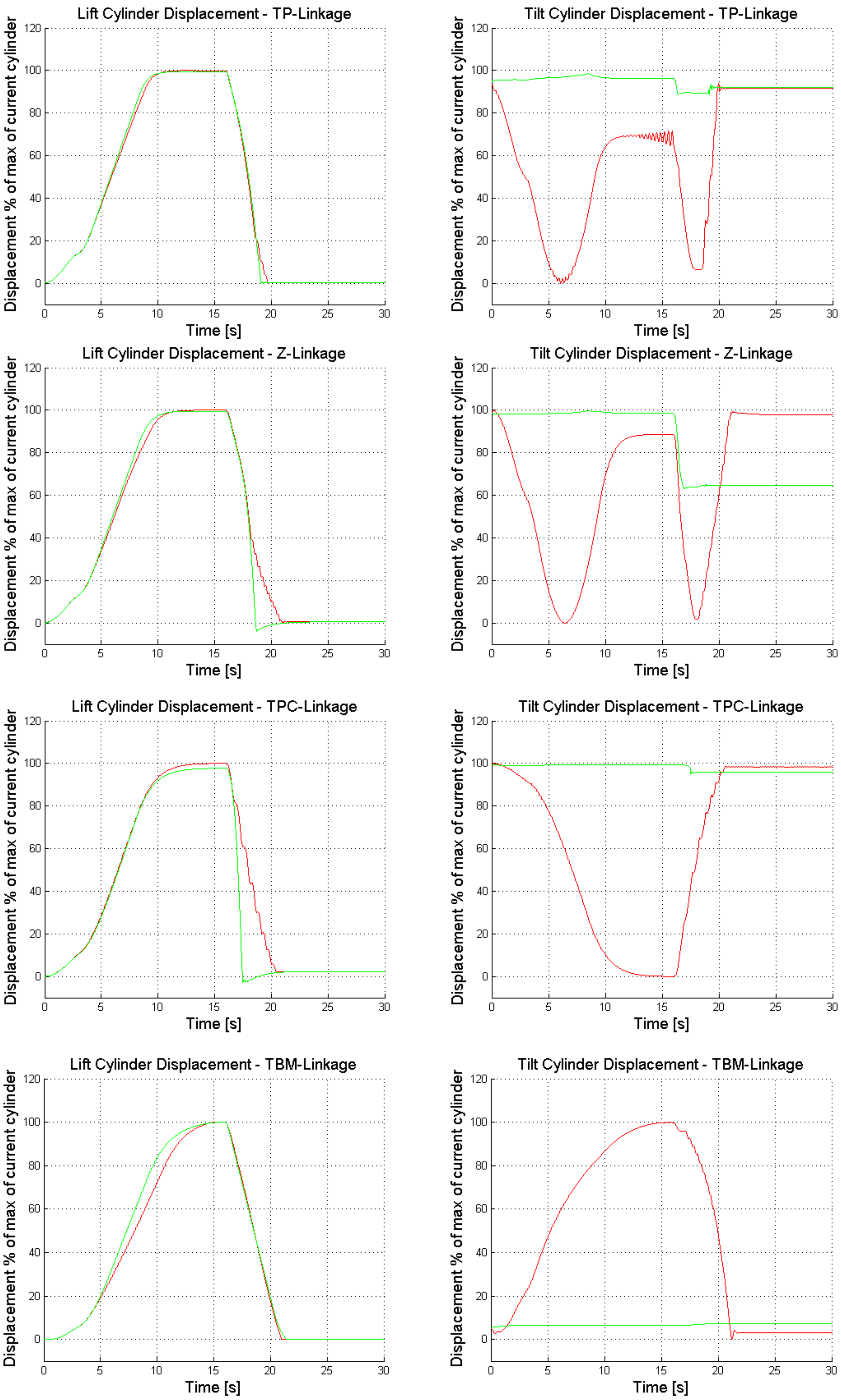

Figure G.2 The curves display the movement of the lift cylinder and the tilt cylinder of the different linkages. The curves are normalized with their own maximum movement, during the fork work cycle with a 6 ton load.

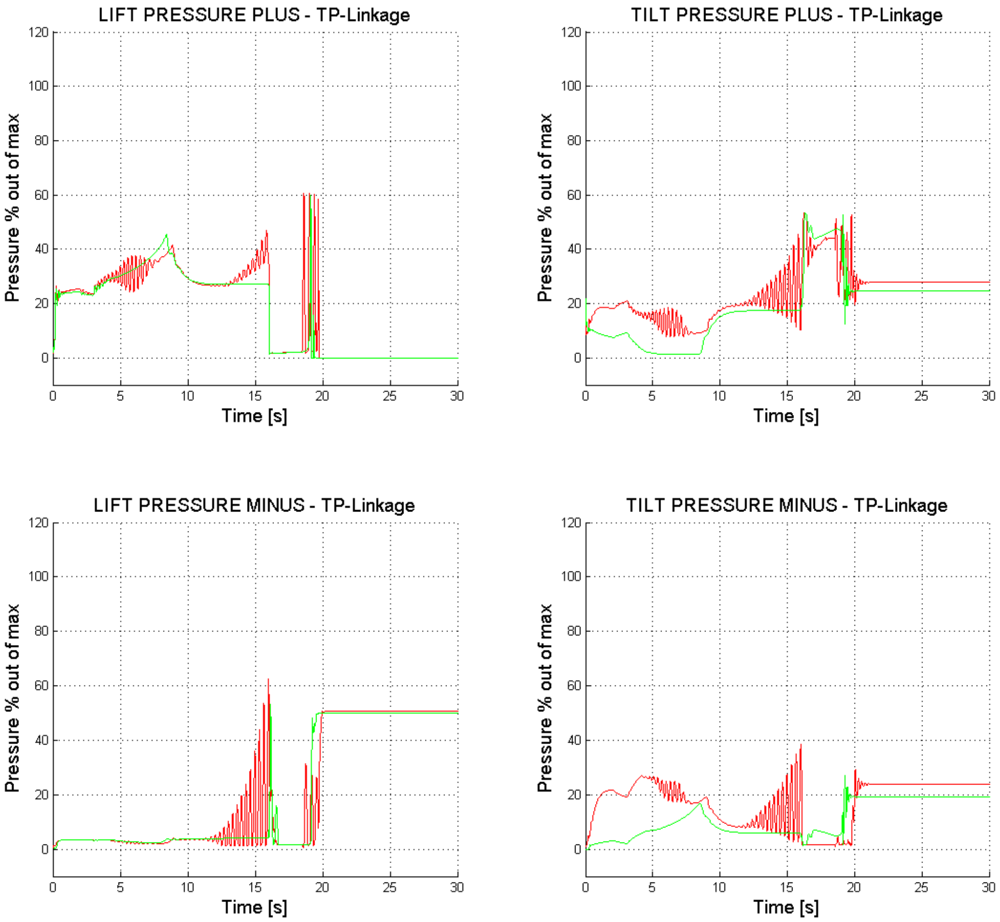

Figure G.3 The pressures on the + and - side of the lift and tilt cylinders for the TP-linkage during the during the fork work cycle with a 6 ton load.

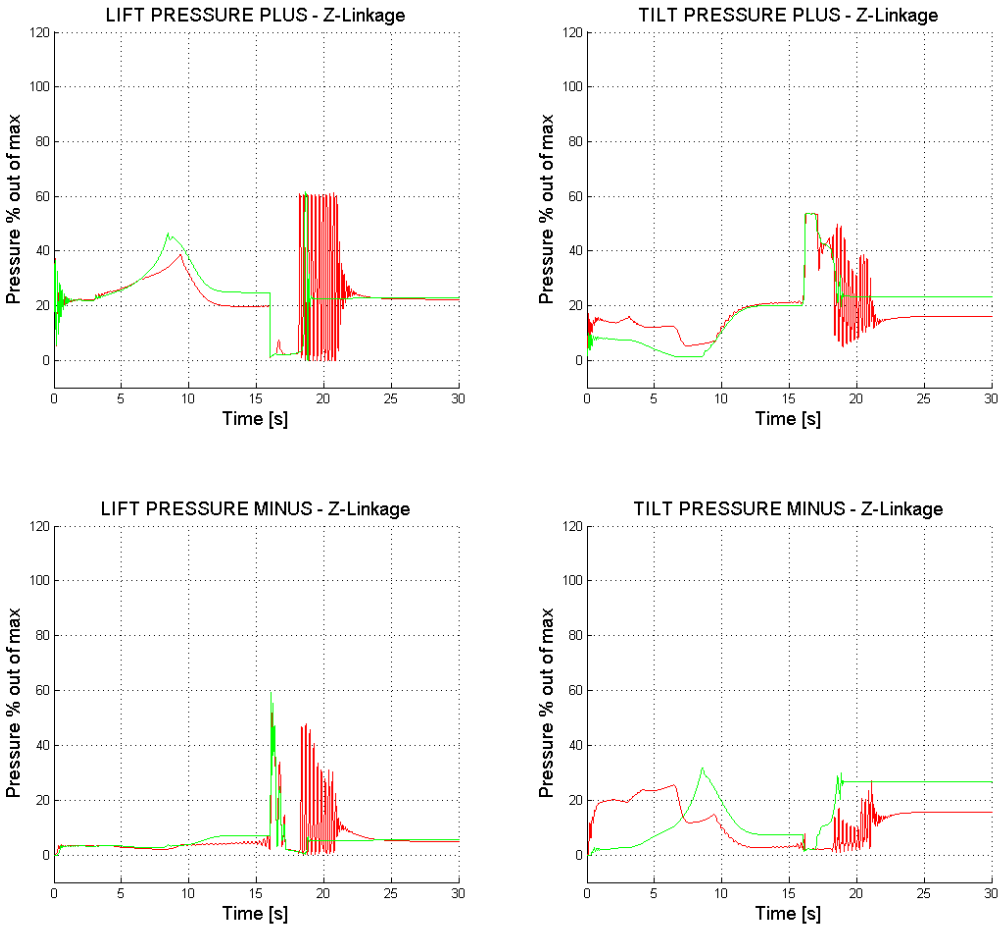

Figure G.4 The pressures on the + and - side of the lift and tilt cylinders for the Z-linkage during the fork work cycle with a 6 ton load.

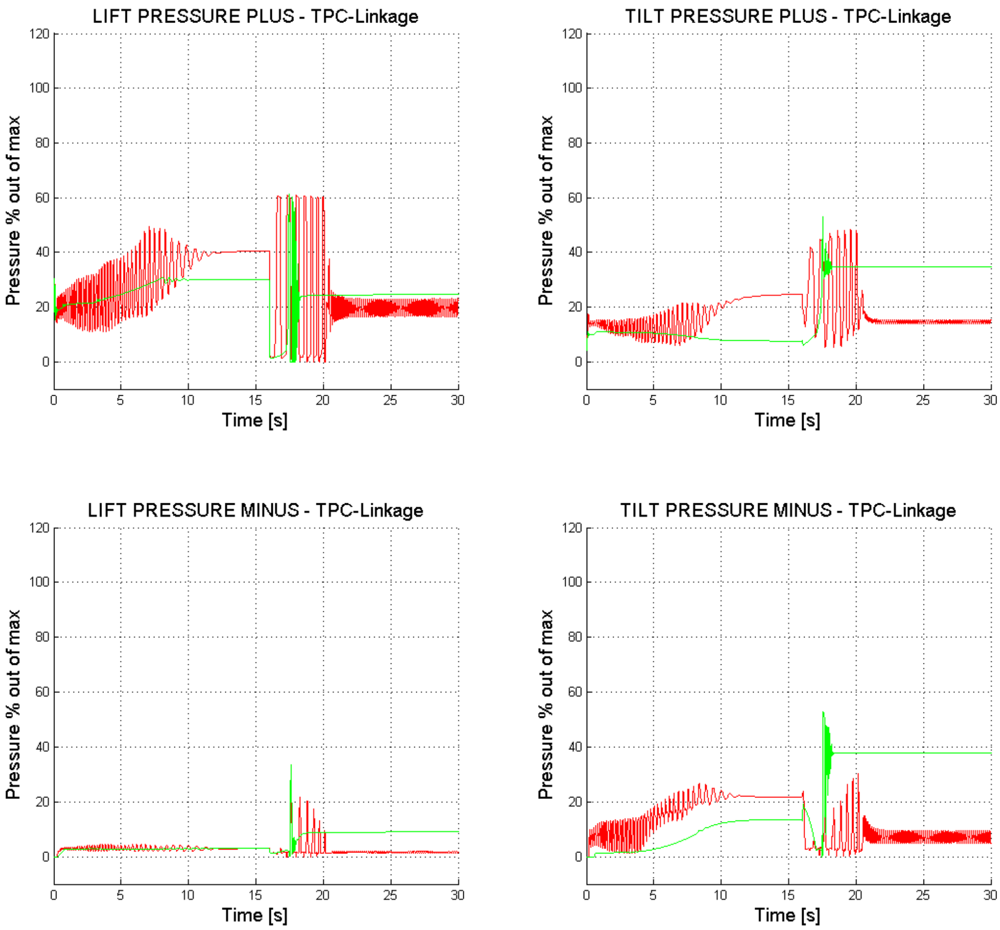

Figure G.5 The pressures on the + and - side of the lift and tilt cylinders for the TPC-linkage during the fork work cycle with a 6 ton load.

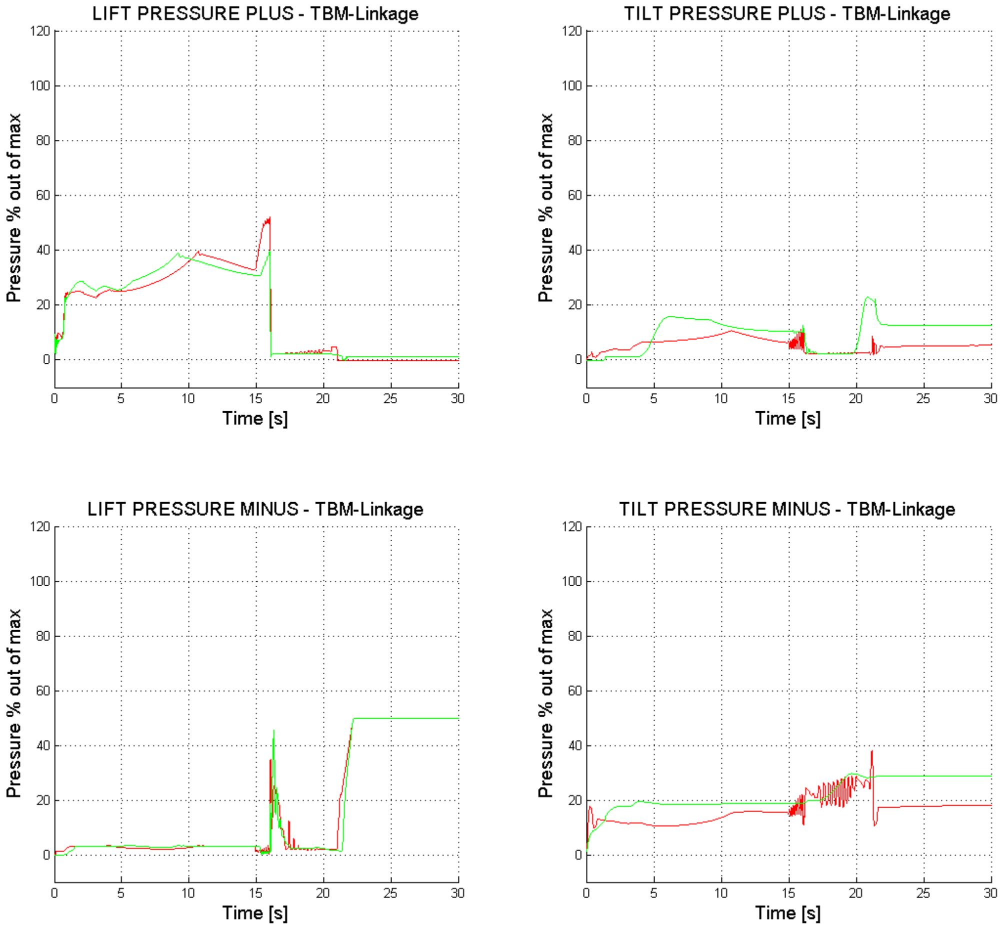

Figure G.6 The pressures on the + and - side of the lift and tilt cylinders for the TBM-linkage during the fork work cycle with a 6 ton load.

Appendix H Cost Result

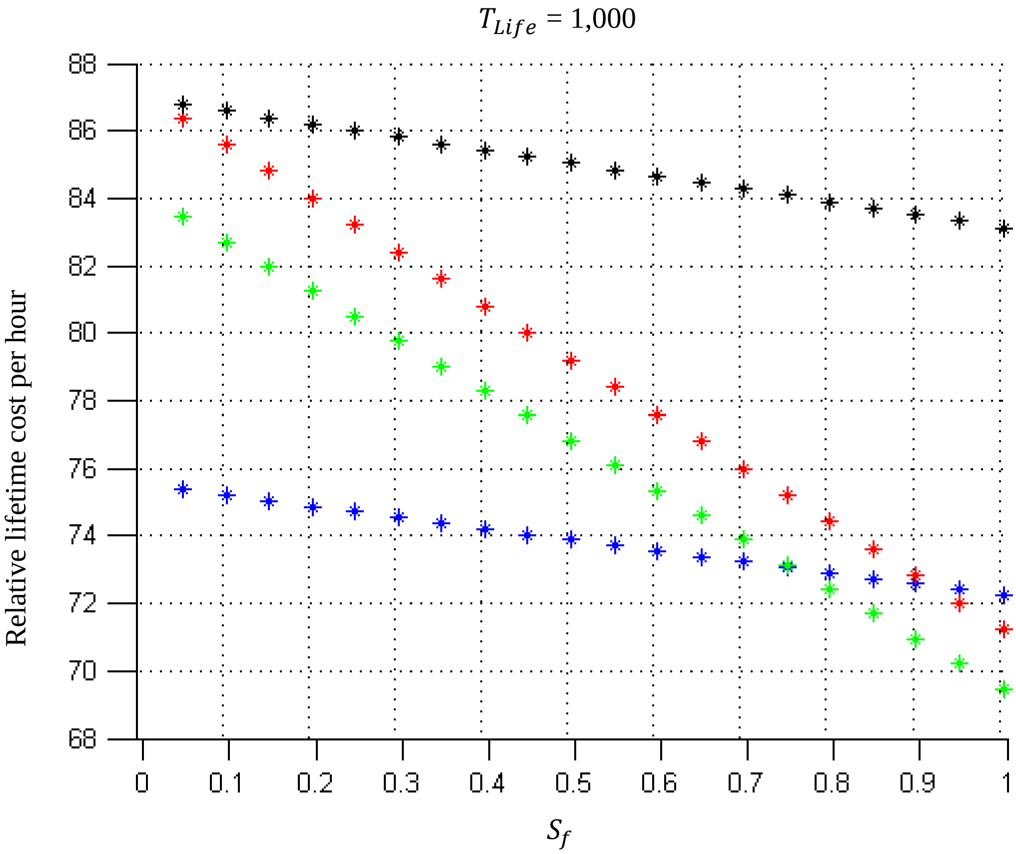

Figure H.1 TCO results from the side where $T_{Life} = 1,000$.

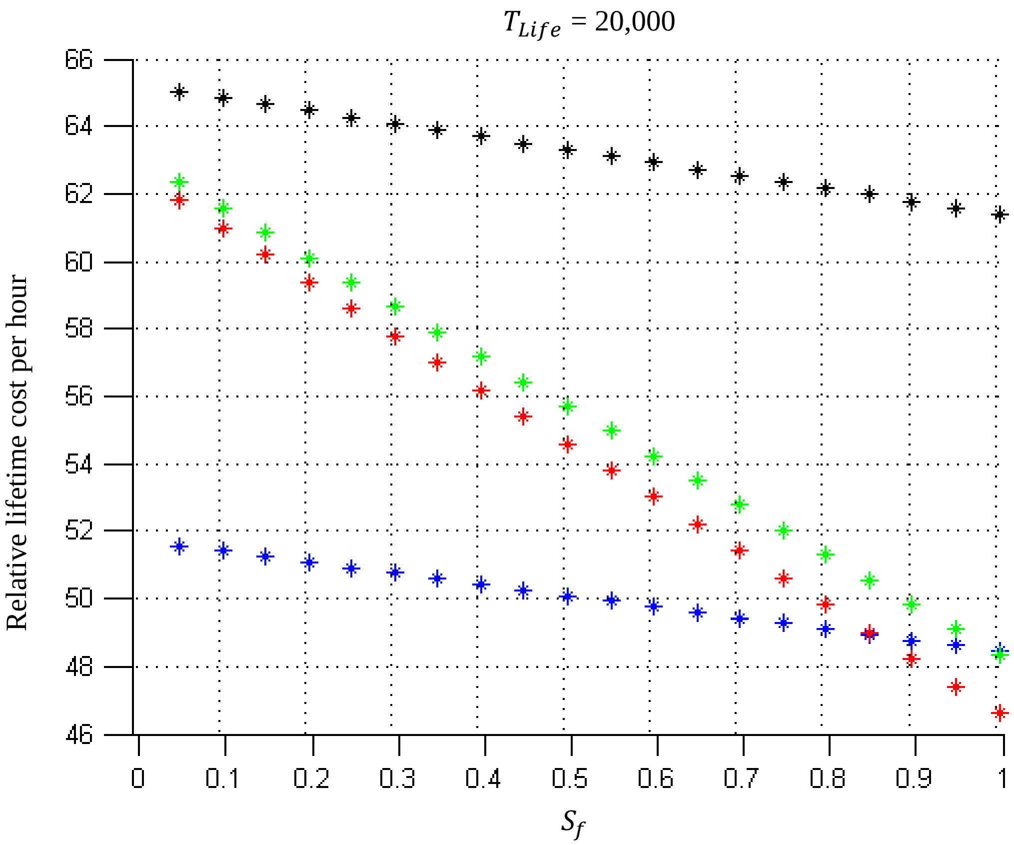

Figure H.2 TCO results from the side where $T_{Life} = 20,000$.

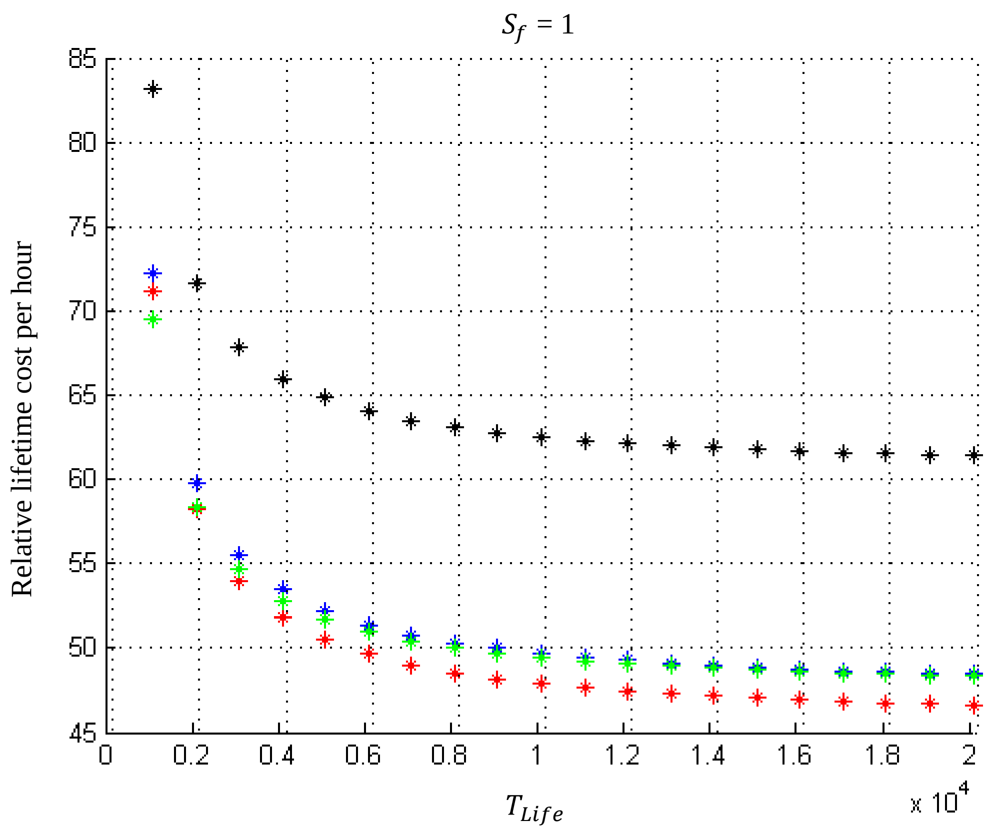

Figure H.4 TCO results from the side where $S_f = 1$.

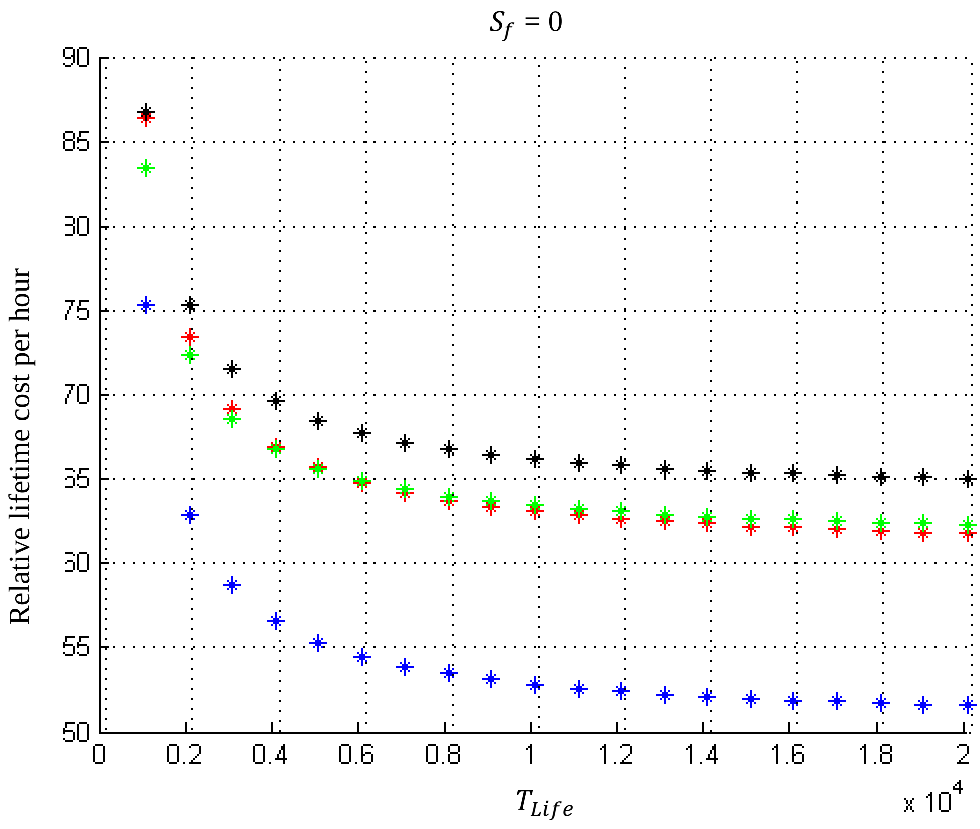

Figure H.3 TCO results from the side where $S_f = 0$.